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MANUAL OF BOILING HEAT-TRANSFER DESIGN CORRELATIONS

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NOTICES

MANUAL OF BOILING HEAT-TRANSFER
DESIGN CORRELATIONS

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ARO, Inc.

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FOREWORD

The survey reported herein was sponsored by Headquarters, Arnold Engineering Development Center (AEDC), Air Force Systems Command (AFSC), Arnold Air Force Station, Tennessee, under Program Element 65401F/063B.

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ABSTRACT

Selected boiling heat-transfer correlations currently being used for design purposes are presented in a graphical form which enables the engineer to compute numerical values of the boiling heat flux parameters rapidly and without knowledge of the physical properties of the fluid other.than at one arbitrary reference state.

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NOMENCLATURE

A	Cross-sectional area
A_S	Surface area
a	Local acceleration
C	Specific heat
C_F	Correction factor
D	Diameter
F	Film boiling parameter
f	Friction factor
g	Acceleration of gravity
g_c	Gravitational constant
h	Specific enthalpy
h	Heat-transfer coefficient
K	Nucleate boiling parameter
k	Thermal conductivity
L	Length
$\dot{m}$	Mass flow rate
N	Dimensionless number
P	Pressure
q/A	Heat flux
T	Temperature
ΔT	Temperature difference
V	Velocity
v	Specific volume
X	Incipient boiling parameter
α	Absorptivity
ϵ	Surface emissivity
η	Constant
λ	Film boiling parameter

μ	Viscosity
ρ	Density
$\Delta\rho$	$(\rho_\ell - \rho_v)$
σ	Surface tension
σ'	Stefan-Boltzmann constant
ϕ	Saturated pool boiling maximum critical heat flux parameter
ψ	Subcooled pool boiling maximum critical heat flux parameter
Ω	Minimum film boiling heat flux parameter

SUBSCRIPTS

BO	Burnout value
bulk	Denotes bulk conditions
c	Convective
crit	Thermodynamic critical conditions
exit	Denotes exit conditions
FC	Forced convection
fg	Change in specific property occurring during transition from liquid to vapor phase
in	Denotes inlet conditions
incip	Denotes conditions at the inception of boiling
ℓ	Liquid phase
MC	Denotes minimum film boiling conditions
max	Maximum
min	Minimum
PR	Prandtl number
RE	Reynolds number
r	Radiative
r	Reduce value
ref	Denotes reference value

sat	Denotes saturation conditions
sub	Denotes subcooled conditions
v	Vapor phase
w	Value at the wall

SECTION I

INTRODUCTION

In designing systems where heat transfer from a solid surface to a fluid is an important consideration, the engineer frequently requires knowledge of the conditions under which the fluid will boil and the magnitude of the heat transfer associated with the boiling process. Considerable research has been devoted toward an understanding of boiling, and numerous correlations are proposed for predicting the inception of boiling and boiling heat-transfer coefficients. However, the problem is extremely complex, and considerable disagreement between the various correlations exists. Moreover, the available experimental data taken under a variety of conditions exhibit so much scatter that it is generally impossible to find data to fit any one of the proposed correlations. Thus, the design engineer is frequently uncertain as to which correlations should be used to obtain the most general prediction of boiling heat transfer.

In addition to this difficulty, boiling heat-transfer correlations are normally expressed in terms of numerous fluid properties raised to fractional powers and thus require a laborious search for property data combined with extensive computational effort for numerical evaluation. For this reason, accompanied with the previously mentioned uncertainty involved in selection of the correct correlation to be used, the engineer often finds it convenient to avoid the boiling heat-transfer problem by overdesigning the system. This, of course, can result in inefficiency or failure of the system to function as desired.

This report attempts to provide a working manual for the design engineer through achievement of the following two specific goals: First, those correlations which in the opinion of the authors give the best agreement with a wide range of reported experimental data have been selected from the boiling heat-transfer literature. Second, these correlations are presented in a form appropriate for expedient evaluation by the engineer.

It is outside the scope of this report to discuss in detail the many theories on the mechanism of boiling and the derivation of the heat-transfer correlations which were reviewed in the literature survey leading up to this writing. The reader interested in these aspects is referred to the excellent surveys already in existence (Refs. 1 through 4). Instead, the present report is divided into five sections, each covering, respectively, one of the following boiling regimes: incipient boiling, fully developed nucleate boiling, maximum critical heat flux, film

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boiling, and minimum critical heat flux. Each section gives an abbreviated description of the boiling regime considered, and the correlation found to have the most general applicability in that regime is stated along with an appropriate reference to which the reader may refer for details of the derivation if desired.

The selected correlation is presented in an effective reduced coordinate system by employing the method of Ref. 5. A brief description of the reduced coordinate system is as follows: Most all boiling correlations investigated were found to reduce to the form

$$f(q/A, \Delta T) = g(P_r)$$

where $f(q/A, \Delta T)$ is a function of the heat flux, q/A, and temperature difference, ΔT, which are the two parameters normally required for design calculations, and $g(P_r)$ is a function of the thermodynamic reduced pressure, $P_r = P/P_{critical}$. If the ratio of the function $f(q/A, \Delta T)$ and $f(q/A, \Delta T)_{ref}$ (where the latter term is the applicable correlation evaluated at an arbitrarily selected reference value of reduced pressure) is plotted against reduced pressure, it forms a single curve for a wide range of different fluids in a manner analogous to the law of corresponding states. With this curve and a table of $f(q/A, \Delta T)_{ref}$ for a number of different fluids, the value of $f(q/A, \Delta T)$ at any given system pressure is computed by a simple multiplication. This technique greatly reduces the work of the design engineer by providing a means of computing a boiling heat flux or corresponding temperature difference without the problem of finding physical properties of the fluid or the tedious numerical computation generally required in evaluating these parameters.

Application of the method is illustrated with a number of example calculations. The final paragraph of each section discusses the accuracy of the correlation with respect to the available experimental data. An estimate of the error associated with the correspondence technique is given and in all cases is small compared with the scatter of experimental data.

Finally, it should be noted that extreme reliability cannot be expected of the prediction technique given herein. This is not a fault of the proposed method but is caused by unreliability of the present day boiling correlations which, unfortunately, because of the complexity of boiling heat transfer, are currently the best available. Presently, greater reliability can only be obtained by experiment on a prototype of the system under design. It is believed, however, that the methods of this manual will predict the required heat-transfer parameters with sufficient accuracy for most practical problems.

SECTION II
INCIPIENT BOILING

Incipient boiling may be defined as follows. Consider a solid, heated surface in contact with a single-phase fluid. Initially the heat transfer from the surface to the fluid takes place by either natural or forced convection. However, as the rate of heat transfer is increased, liquid next to the surface eventually vaporizes, forming small vapor bubbles. The first appearance of these bubbles is called incipient boiling, and the heat flux and wall temperature associated with this condition are called the incipient boiling heat flux, $(q/A)_{incip}$, and the incipient boiling wall temperature, $(T_w)_{incip}$.

With the inception of boiling, the rate of heat transfer departs from that predicted by standard convection correlations. Accordingly, the value of $(q/A)_{incip}$ serves as a criterion to predict the beginning of the transition from purely convective heat transfer to nucleate boiling heat transfer. For engineering purposes, $(q/A)_{incip}$ is analogous to the critical Reynolds number which distinguishes the laminar flow regime from the turbulent flow regime. The magnitude of $(q/A)_{incip}$ determines whether a system should be designed on the basis of purely convective heat-transfer calculations or on the basis of boiling heat-transfer calculations.

The incipient boiling heat flux, $(q/A)_{incip}$, is also of interest to designers of cryogenic flow measuring equipment or fuel lines where internal vapor formation must be avoided. In these systems, $(q/A)_{incip}$ represents the maximum permissible heat leak. It is thus desirable to have means of evaluating the magnitude of $(q/A)_{incip}$ and also, the corresponding $(T_w)_{incip}$.

To this end, one frequently predicts the start of boiling based on the criterion that boiling begins when either (1) the bulk fluid temperature reaches the saturation value or (2) the system wall temperature reaches the saturation value. Although the latter method may give realistic approximations, neither criterion is exactly correct. Experiments have demonstrated, in fact, that boiling will occur on heated surfaces in fluids with bulk temperatures many degrees below saturation temperature. Although bulk temperature will subsequently be shown to have a strong influence on incipient boiling conditions, the relationship $T_{bulk} = T_{sat}$ fails as a criterion for predicting the beginning of boiling.

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In turn, the assumption that boiling begins at $T_w = T_{sat}$, although reliable in some cases, may be badly in error in others. Many experiments, particularly those conducted at high flow rates and low system pressures, have shown that the wall temperature may exceed the saturation temperature by several degrees before the first vapor bubbles are generated, and hence, boiling begins.

Since the aforementioned criteria are not well suited for predicting the inception of boiling, a more reliable method is needed. The purpose of the following section is to present such a method.

The proposed method is valid for both natural and forced flow systems where the heat transfer to the fluid is from a commercially finished surface. A commercially finished surface refers to one formed by standard industrial production procedures such as extruding, grinding, rolling, etc. The method is not applicable to extremely smooth surfaces, i. e., highly polished or glass surfaces. Inasmuch as industrial surfaces are, in general, produced by standard production processes, the above limitation is not considered severe. As a matter of interest, however, should the method be used to predict incipient boiling on an extremely smooth surface, the predicted $(T_w)_{incip}$ will be low and in this sense conservative.

Attention is now directed to Fig. 1 (Appendix I) and Table I (Appendix II) which may be used to predict the incipient boiling conditions provided the system pressure and Prandtl number of the fluid are known. Before a detailed description of the procedure is given, however, Fig. 1 and Table I will be described.

Figure 1 is a plot of X/X_{ref} versus reduced pressure. The X represents the value of $\left(\Delta T_{sat}/\sqrt{q/A} \, NPR\right)_{incip}$ computed at a specific value of reduced pressure, and X_{ref} is the same quantity evaluated at an arbitrarily selected reference reduced pressure of 0.05. From a known value of X, the relationship between $(q/A)_{incip}$ and $(\Delta T_{sat})_{incip}$ is given by

$$(q/A)_{incip} = (\Delta T_{sat})^2_{incip}/N^2_{PR}X^2 \tag{1}$$

Table I lists values of X_{ref} for a number of fluids. Evaluation of X_{ref} for other fluids is possible through the equation

$$\left(\frac{\Delta T_{sat}}{N_{PR}\sqrt{q/A}}\right)_{incip} = \sqrt{\frac{8\sigma v_{fg}T_{sat}}{h_{fg}k\mu}} \tag{2}$$

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where the fluid properties are determined at the saturation temperature corresponding to a reduced pressure of 0.05. Since X appears to obey the principle of corresponding states, its value can be determined at any desired pressure from Fig. 1 once X_{ref} is known. This statement does not apply to nonwetting fluids.

The mathematical analysis and physical model from which Eq. (1) is derived are discussed in Ref. 6. Since the purpose of this report is to supply information on boiling from the design point of view, no further discussion of the theory will be given.

With this knowledge of Fig. 1 and Table I, we can now turn to the chief steps involved in finding the condition of heat flux and temperature at incipient boiling. These are as follows:

1. The design pressure for the system is determined.

2. The reduced pressure, P_r, is calculated.

3. From Fig. 1, X/X_{ref} corresponding to the calculated value of P_r is evaluated.

4. From Table I, X_{ref} for the fluid of interest is selected.

5. The product of (X/X_{ref}) and X_{ref} determines X $\left(X = (\Delta T / \sqrt{q/A} \, N_{PR})_{incip}\right)$.

6. The relationship between the incipient boiling heat flux and $(\Delta T_{sat})_{incip}$ is thus found to be $(q/A)_{incip} = (\Delta T_{sat})^2_{incip} / X^2 N^2_{PR}$, where N_{PR} is determined from Table II at the value of reduced pressure calculated in step 2.

Steps 2 through 5 need no further discussion; however, comments on steps 1 and 6 are necessary.

The appropriate value of system pressure to be used in a given calculation is dependent on whether the system is static or dynamic. For a static system, such as a storage tank, boiling will occur first where the pressure is a maximum, other conditions being equal. In this case the pressure for use in step 1 is evaluated where the hydrostatic head is the greatest.

Where the fluid is in motion, the point of incipient boiling is governed by both the local pressure and bulk fluid temperature. The influence of T_{bulk} will be illustrated subsequently as having, in general, a more pronounced effect on the start of boiling than pressure, and hence, the

pressure to be used should correspond to the local value where T_{bulk} is a maximum.

Turning to the results of step 6, the relationship

$$q/A = (\Delta T_{sat})^2_{incip}/X^2N^2PR \tag{1}$$

can be used to solve three types of problems: (1) those where q/A is known and the value of T_w at which boiling will begin is to be determined, (2) those where the wall temperature is specified and it is desired to know the maximum heat flux which can be transferred without boiling, and (3) those where neither q/A nor T_w is fixed, but the single-phase heat-transfer characteristics are known. In the latter-type problem, Eq. (2) is modified as follows. The heat flux is written

$$q/A = h(T_w - T_{bulk}) \tag{3}$$

where h is the standard single-phase heat-transfer coefficient applicable to the geometry of interest. The addition and subtraction of T_{sat} within the brackets permit writing.

$$q/A = h[(T_w - T_{sat}) - (T_{bulk} - T_{sat})] \tag{4}$$

where

$$\Delta T_{sub} = T_{sat} - T_{bulk}$$

which gives

$$q/A = h(\Delta T_{sat} + \Delta T_{sub}) \tag{5}$$

Introducing this form of q/A into Eq. (1) and rearranging gives

$$(\Delta T_{sat})^2 - X^2N^2PRh(\Delta T_{sat}) - X^2N^2PRh\Delta T_{sub} = 0 \tag{6}$$

the roots of which are

$$\Delta T_{sat} = \frac{X^2N^2PRh}{2} \pm \sqrt{\frac{(X^2N^2PRh)^2}{4} + X^2N^2PRh\Delta T_{sub}} \tag{7}$$

The minus sign, of course, has no physical meaning.

The incipient boiling wall superheat is thus

$$(\Delta T_{sat})_{incip} = (T_w)_{incip} - T_{sat} = \frac{X^2N^2PRh}{2} + \sqrt{\frac{(X^2N^2PRh)^2}{4} + X^2N^2PRh\Delta T_{sub}} \tag{8}$$

Once $(T_w)_{incip}$ is known, $(q/A)_{incip}$ can be calculated from Eq. (3).

The procedure for predicting incipient boiling outlined above is now illustrated by two example problems.

Example 1: Suppose it is desired to predict the incipient boiling heat flux on a surface having a fixed temperature, $T_w = 180°R$. Let the fluid be liquid nitrogen at a pressure of 50 psia.

The first step is to calculate P_r. From tabulated data, P_{crit} is found to equal 493 psia, and hence, $P_r = P/P_{crit} = 0.1$. At a reduced pressure of 0.1, X/X_{ref} is found from Fig. 1 to have a value of 0.73 and from Table I, X_{ref} for liquid

nitrogen is given as $0.017°R/\left[Btu/hr\text{-}ft^2\right]^{1/2}$. Taking the product of (X/X_{ref}) and X_{ref} gives

$$\left(\frac{\Delta T_{sat}}{N_{PR}\sqrt{q/A}}\right)_{incip} = \left(\frac{X}{X_{ref}}\right) X_{ref}$$

$$= (0.73)(0.017)$$

$$= 0.012°R/[Btu/hr\text{-}ft^2]^{\frac{1}{2}} \qquad (9)$$

Transposing

$$7. \left(\frac{1}{.012}\right)^2$$

$$(q/A)_{incip} = (83.1)^2 \left(\frac{\Delta T_{sat}}{N_{PR}}\right)^2 \qquad (10)$$

The value of ΔT_{sat} is calculated from the wall temperature, $T_w = 180°R$, which is given, and from the saturation temperature, $T_{sat} = 161°R$, which is found from thermodynamic tables for liquid nitrogen. The Prandtl number, $N_{PR} = 2.1$, corresponding to $P_r = 0.1$, is found from Table II. Substitution of these values into Eq. (10) gives the required heat flux

$$q/A = 565,000 \ Btu/hr\text{-}ft^2$$

The preceding simple problem, although useful in illustrating the prediction technique embodied in Fig. 1 and Table I, is not as frequently encountered in practical applications as the following more complex problem where neither T_w nor q/A is known a priori.

Example 2: Liquid nitrogen flows through a 100-ft-long, 0.5-in.-ID pipe at a rate of 1.0 lb_m/sec. The fluid is required to enter the pipe at a pressure of 100 psia and a bulk

temperature of 100°R and to leave at 140°R. It is desired to determine the maximum tube wall temperature which can be obtained before the fluid will boil.

As neither T_w nor q/A is known in advance, the calculation begins with Eq. (8).

$$(\Delta T_{sat})_{incip} = \frac{h(N_{PR}X)^2}{2} + \sqrt{\frac{[(N_{PR}X)^2 h]^2}{4} + (N_{PR}X)^2 h \Delta T_{sub}} \qquad (8)$$

Solving this equation for ΔT_{sat} allows us to find the wall temperature which will cause boiling from the relationship

$$(T_w)_{incip} = T_{sat} + (\Delta T_{sat})_{incip}. \qquad (11)$$

Inspection of Eq. (8) reveals that $(\Delta T_{sat})_{incip}$ will vary locally with X, through its variation with pressure, and also with subcooling as ΔT_{sub} appears under the radical. Since X has its lowest value at the inlet (highest pressure) where ΔT_{sub} has its largest value, the reverse being true at the exit, the question arises as to whether inlet, exit, or average values of the parameters entering Eq. (8) should be employed to evaluate $(\Delta T_{sat})_{incip}$. In order to answer this question, a preview of the final solution to our example problem is given in Table III.

Table III shows the value of the incipient boiling wall temperature computed on the basis of inlet conditions, average conditions, and exit conditions, respectively. Also, in order to obtain at the same time a comparison of the boiling characteristic of a cryogenic with a more commonly known fluid, such as water, calculated results for water in a similar flow system are also tabulated.

One notes in either case that the lowest value of wall temperature which precipitates boiling occurs at the exit and the highest value at the entrance. In general, such conditions being true suggest that conservative design of a system to operate in the nonboiling regime should be based on calculations at exit conditions. The opposite is true for conservative design of systems to operate only in the boiling heat-transfer regime.

Now, continuing the calculation of the example problem, we will compute the solution using exit parameters since our

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purpose is to determine the maximum wall temperature the tube could obtain without boiling.

The value of the exit pressure is computed from

$$\Delta P = P_{in} - P_{exit} = f \frac{8L\dot{m}^2}{\pi^2 g_c \rho D^5}$$

where the friction factor, f, can be determined from

$$f = \frac{0.316}{N_{RE}^{\frac{1}{4}}}$$

Thus

$$P_{exit} = 61 \text{ psia}$$

The reduced pressure is then $P_r = 0.12$. From Fig. 1 and Table II, X/X_{ref} and N_{PR} are, respectively, 0.65 and 2.05. Table I gives a value of 0.017 for X_{ref}, and on multiplication, X becomes 0.011.

The parameter, h, in Eq. (8) refers to the single-phase heat-transfer coefficient which can be determined from

$$h = 0.023 \left(\frac{k}{D}\right)_{bulk} (N_{RE})_{bulk}^{0.8} (N_{PR})_{bulk}^{0.4} \qquad (12)$$

Although the Dittus-Boelter forced convection heat-transfer correlation is used here, any other standard correlation applicable to the given geometry and flow conditions could be equally well employed.

Attention is drawn to the "bulk" subscripts, which indicate the properties are evaluated at bulk temperature (local value). A word of caution is given against mistakenly substituting in Eq. (6) the value for N_{PR} determined in the preceding step ($N_{PR} = 2.05$) which is based on saturation temperature. The correct value of N_{PR} needed for calculating h is 3.27, based on bulk temperature.

Evaluating Eq. (6) gives

$$h = 2863 \text{ Btu/hr-ft}^2\text{-}^\circ F$$

With the magnitudes of $X = 0.011$, $N_{PR} = 2.05$, and $h = 2863$ Btu/hr-ft^2-$^\circ$F known, substitution into Eq. (8) gives the incipient boiling superheat as

$$(\Delta T_{sat})_{incip} = 4.4^\circ R$$

The maximum value of the wall temperature that the tube can have without boiling is thus 168°R.

Knowledge of the corresponding heat flux is immediately obtained from Eq. (1) and is

$$(q/A)_{incip} = 37{,}900 \text{ Btu/hr-ft}^2$$

Having now established a method for predicting thermal conditions at incipient boiling, we turn to a consideration of the accuracy of our results. Figure 2 compares the curve of Fig. 1 with experimental data taken from the literature. Unfortunately, there is only a limited number of experiments in which the incipient boiling point has been carefully reported. Data from these are represented by the shaded circles on Fig. 2 and are for water, neon, and carbon tetrachloride (Refs. 7, 8, and 9, respectively). The remaining data (those which are enclosed by the shaded area) were taken from reports where only approximate values of the incipient boiling point could be obtained. The nature of the approximation was such that the data will invariably be higher than the theory predicts. For this reason, the fact that the curve lies along the lower limits of the shaded region is not unexpected. Although the majority of the data are not accurate, they do indicate that the trend of the experimental finding is correctly predicted by the theory.

If comparison of the theory is restricted to the most reliable data, those given by the shaded circles, an accuracy of ±40 percent is indicated. Greater accuracy than this cannot be expected from the proposed method because the effects of heater surface finish are not taken into account. Despite this limitation, however, the method proposed here for predicting incipient boiling is believed to be the best currently available for design purpose. This is particularly true in view of the fact that the correlation (Eq. (2)) can be plotted in terms of reduced coordinates as shown. Direct evaluation of Eq. (2) is thus eliminated through this plot, and hence, the time required for both numerical computation and for property data evaluation is greatly reduced.

Of course, there is some inaccuracy incurred by using the principle of corresponding states (that is, from assuming Eq. (2) for all fluids can be represented by one curve). This is small in comparison to the experimental scatter as shown by the error bands (dashed lines) drawn in Fig. 2. These error bands were obtained by calculating directly from Eq. (2) values for X/X_{ref} for seven different fluids. The average value at a given P_r was then taken and the standard deviation of the calculated values from the average determined. Two standard deviations were plotted and constitute the error bands shown in Fig. 2. The

error generated by presenting the incipient boiling correlation on reduced coordinates is obviously negligible compared with the uncertainty involved between the experimental results and the theory.

SECTION III
FULLY DEVELOPED NUCLEATE BOILING

Nucleate boiling occurs on a heated surface when the heat-transfer rate exceeds that which can be removed by single-phase convection. Vapor then forms as bubbles at specific nuclei on the surface. In a saturated fluid the bubbles leave the surface carrying the vapor to a free interface where it may escape. This form of boiling generally occurs in a natural flow system; it can, however, occur in a forced flow system. In subcooled fluids, those where the bulk temperature is less than the saturation temperature, vapor bubbles grow only in a superheated layer of liquid next to the heated surface. Once protruding from this layer, the bubbles contact the colder fluid and immediately collapse. Boiling of this type is called subcooled or local boiling and occurs most frequently in forced flows.

Although it is known that extremely high rates of energy removal at modest wall temperatures are obtained when a fluid boils on a heated surface, the exact mechanism of energy transfer is not completely understood. The high turbulence and latent heat transfer associated with the growth and motion of vapor bubbles, however, are generally agreed as being the dominant factors influencing the energy exchange. Experimentally, bubble formation is shown to be characterized by the difference between the heated wall temperature and the saturation fluid temperature, ΔT_{sat}, such that in general

$$q/A \propto \Delta T_{sat}^{n}$$

The value of n is on the order of three for practical conditions. Interestingly, bulk fluid temperature does not directly influence the thermal driving potential, nor does velocity-induced turbulence significantly affect the heat-transfer rate. As a consequence, saturated boiling with natural convection and subcooled boiling with forced convection may be reasonably correlated with the same equation. This statement does not apply to saturated bulk boiling in a forced flow system, which will be discussed in a later report.

The engineer frequently encounters nucleate boiling problems in designing cooling systems for high energy devices such as rocket nozzles, exhaust diffusers, etc. Normally, subcooled boiling is important

in these applications. Both subcooled boiling and saturated boiling also occur readily in equipment where cryogenic fluids are employed. Since the design engineer must predict piping dimensions, type of fluids, mass flow rates, and other parameters for these systems, a method of predicting the nucleate boiling heat flux is required.

Information for predicting the needed nucleate boiling heat-transfer parameters is contained in Fig. 3 and Table IV. The correlation given is applicable to saturated and subcooled boiling with the restriction that results predicted for saturated fluids pertain only to natural flow systems.

Figure 1 is a plot of a boiling correlation proposed by Kutataladze (Ref. 10).

$$K = \frac{\Delta T_{sat}}{(q/A)^{0.3}} = \frac{1429}{k_\ell} \left[\frac{\sigma}{(\rho_\ell - \rho_v)} \right]^{0.5} \left[\frac{\mu \rho_v^2 h_{fg}(\rho_\ell - \rho_v)}{P_{sat}\rho_\ell\rho_v} \right]^{0.7} N_{PR}^{0.35} \quad (13)$$

The abscissa is reduced pressure, and the ordinate is the ratio of K/K_{ref}, where K is computed from Eq. (13) evaluated at any given reduced pressure and K_{ref} is computed from Eq. (13) evaluated at a reference reduced pressure of 0.05.

Although numerous other boiling heat-transfer correlations are available, Kutataladze's was selected for use here since it appears to agree best with reported experimental data over a wide range of pressures for a large variety of fluids. Dimensional analysis was employed in determining Eq. (13), for which the details may be found in Ref. 10. No account for surface finish is contained in Kutataladze's expression. Moreover, inasmuch as the empirical constants were obtained from experimental data taken for wetting fluids on commercially finished surfaces, the predicted results from Eq. (13) are only valid for such conditions. For nonwetting fluids or for extremely smooth surfaces, the prediction technique described herein is subject to error.

Table IV lists values of K_{ref} for several different fluids. If K_{ref} is desired for a fluid not listed, it may be calculated from Eq. (13) using property values determined at T_{sat} corresponding to a reduced pressure of 0.05. With K_{ref} thus determined, K at any other reduced pressure is found immediately from Fig. 3.

Attention is now given to the steps followed in determining the relationship between q/A and ΔT_{sat} from Fig. 3 and Table IV. These are:

1. Determine the system design pressure.

2. Calculate the reduced pressure, P_r.

3. Find K/K_{ref} from Fig. 3 at the calculated value of P_r.

4. Select K_{ref} for the fluid of interest from Table IV.

5. Multiply K/K_{ref} by K_{ref} to obtain

$$K = \Delta T_{sat}/(q/A)^{0.3}$$

6. Transpose and obtain the required expression

$$q/A = (\Delta T_{sat}/K)^{3.33}$$

In most analyses, the use of an overall average system pressure in step 1 provides accuracy consistent with the prediction technique given. However, if extreme pressure gradients are anticipated in the flow system under design, then the flow path should be subdivided into sections and the analysis carried out in steps based on pressure averaged over each section. The pressure drop between sections may be approximately calculated from single-phase pressure drop relationships provided the flow is everywhere highly subcooled, $\Delta T_{sub} \geq 50°F$, and the diameter or characteristic dimension of the flow channel is large compared to the thickness of the vapor formation, $D \geq 0.25$ in. Pressure loss calculations for flows outside these limits will be discussed in a later report.

From the result of step 6

$$q/A = (\Delta T_{sat}/K)^{3.33} \tag{14}$$

the heat flux removable from a wall at a fixed temperature can be found. Of course, the reverse problem of finding the wall temperature at a given heat flux is also solvable.

If one prefers to work with a heat-transfer coefficient, then

$$h = \frac{q/A}{\Delta T_{sat}} = \frac{\Delta T_{sat}^{2.33}}{K^{3.33}}$$

Generally, with boiling heat transfer, h is based on ΔT_{sat} as shown.

The following example problems serve to illustrate the application of the above outlined procedure for calculating boiling heat transfer.

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Example 1.   Liquid nitrogen under a pressure of 50 psia boils on a heated surface.   Find the heat flux at a wall temperature of 180°R.

The critical pressure of liquid nitrogen is found as P_{crit} = 493 psia from published data.   The reduced pressure is

$$P_r = 50/493 = 0.101$$

From Fig. 3

$$K/K_{ref} = 0.76$$

From Table IV

$$K_{ref} = 0.913$$

Then

$$K = (K/K_{ref}) \, K_{ref}$$
$$= (0.76) \, (0.913)$$
$$= 0.690$$

From Eq. (14)

$$q/A = (\Delta T_{sat}/0.690)^{3.33}$$

At T_w = 180°R and T_{sat} = 161°R

$$\Delta T_{sat} = 19°R$$

and

$$q/A = (27.5)^{3.33}$$

To facilitate the calculation of q/A, Fig. 4 has been prepared. From this, the value of $(27.5)^{3.33}$ is found to be 6.5 x 10^4. Hence

$$q/A = 6.5 \times 10^4 \; Btu/hr\text{-}ft^2$$

Example 2.   The rate at which energy must be removed from a rocket nozzle diffuser is 2 x 10^6 Btu/hr.   The diffuser walls are encased by an insulated outer wall, and cooling water flows through the enclosure.   Estimate the surface area required in contact with the fluid if the wall temperature of the diffuser is to be maintained below 400°F.   The average system pressure is 50 psia.

Assuming the fluid undergoes subcooled boiling and neglecting entrance and exit effects, the surface area is found by re-arranging Eq. (14) to give

$$A_s = q/(\Delta T_{sat}/K)^{3.33}$$

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The reduced pressure of water at 50 psia is $P_r = 0.016$, and from Fig. 3, K/K_{ref} is determined as 1.5. Hence

$$K = (K/K_{ref})\ K_{ref} = (1.5)\ (0.628)$$

where $K_{ref} = 0.628$ is taken from Table IV

$$K = 0.945$$

The saturation temperature corresponding to $P = 50$ psia is 281°F and with $T_w = 400°F$, ΔT_{sat} becomes 119°F. Thus

$$A_s = \frac{2 \times 10^6\ Btu/hr}{(126)^{3.33}\ Btu/hr\text{-}ft^2}$$

The value of $(126)^{3.33}$ is conveniently found from Fig. 4 as 10^7. The required surface area is

$$A_s = 0.20\ ft^2$$

It is interesting to compare this result with the area determined from a calculation based on a forced convection heat-transfer coefficient, rather than on a boiling coefficient. From Ref. 11 one finds that h for forced convection with water is on the order of 2000 Btu/hr-ft². Thus, for our problem

$$A_s = \frac{q}{h(T_w - T_{bulk})}$$

$$= \frac{10^3}{T_w - T_{bulk}}$$

The maximum temperature difference one could expect is on the order of 450°F. Hence

$$A_s = 2.2\ ft^2$$

Obviously the cooling system would be oversize if designed on the assumption of purely convective heat exchange.

An indication of the accuracy expected from the proposed prediction technique is given in Fig. 5. Experimental data from the literature are compared with the theoretical correlation. The outlined area represents the limits within which data for nine different fluids undergoing natural or forced convection boiling lie. The forced convection data include results reported for flow in cylindrical, annular, and rectangular ducts and for flow across tubes. The range of fluid velocities covered is from 0 to 21 ft/sec. It should be noted that the majority of the forced flow boiling data available is for water.

Considerable discrepancy between the theory and the data is apparent. This is largely because of appreciable scatter in the experimental data resulting from heater surface finish effects which are difficult to take into account. Unfortunately, until the influence of this liquid-solid interface on the boiling phenomenon is understood, it is doubtful that a more accurate general purpose correlation than that given here will be possible. However, if one remains cognizant of the involved uncertainty, results predicted by Kutataladze's boiling correlation should prove useful for many general problems in the design of boiling systems. This is particularly true when the boiling correlation is given in terms of reduced coordinates, as done here. The evaluation of Eq. (13) then requires very little time or effort, and the results can be expected with 95-percent confidence to deviate no more than the shaded error bands in Fig. 5 show. Obviously this error is negligible compared with the uncertainty in the correlation.

SECTION IV
PEAK HEAT FLUX

The peak heat flux* is defined as that value of the heat flux at which the boiling process undergoes a transition from nucleate boiling (described in Section III) to film boiling (described in Section V). This transition occurs when the mechanism by which vapor is removed from the heated surface during nucleate boiling becomes unstable, breaks down, and a film of vapor blankets the heater. The formation of a vapor film imposes a high thermal resistance to heat transfer from the surface, and if the surface is heated at a constant rate, the wall temperature rises rapidly in order to maintain the constant flow of heat to the coolant. For boiling fluid having a high saturation temperature such as water, the melting temperature of the heated material is exceeded, and the structure fails. For fluids which boil at lower temperatures, such as cryogenics, the rise in temperature of the solid surface accompanying the formation of the vapor film is not, normally, sufficient to melt the surface, and heat exchange continues by the mechanism of film boiling — however, at a considerable reduced efficiency.

The transition to film boiling is much less abrupt when the surface undergoing boiling is heated by a convection process rather than by a constant heat generation process (e. g., a tube heat exchanger where boiling takes place on the outer surface of the tubes because of a high

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*Also referred to as the critical heat flux or burnout heat flux.

temperature fluid flowing internally). In this case, when the film forms, the heat flux decreases gradually with the rising wall temperature, and the boiling process passes through a partially nucleate-partially film boiling regime in which the heat flux continues to decrease with increasing ΔT_{sat}. This form of boiling proceeds until the vapor film stabilizes, at which time the heat flux once more rises with increasing temperature potential. The heat flux at which the film becomes stable is referred to as the minimum film boiling heat flux, second critical heat flux, and other such names (see Section VI). With convection surface heating, sudden melting of the structure does not dramatically occur as it does with high, constant surface heating.

In view of the above, it is apparent that a means of predicting the peak heat flux is a very pressing problem in the design of cooling systems for constant output, high energy devices such as rocket nozzles, diffusers, or nuclear reactors which must operate below the peak heat flux if structural integrity is to be maintained.

Knowledge of the peak heat flux, although not as pressing, is also important in low energy systems such as cryopumps or cryogenic storage vessels which operate safely either under conditions of film boiling or nucleate boiling. Here, however, a prediction of the peak heat flux is necessary to determine the appropriate boiling correlation (film or nucleate) for design purposes.

Considerable controversy exists over the most realistic physical model from which a method for predicting burnout can be developed. However, it is known from experiments that the major parameters which influence the peak heat flux are pressure, fluid velocity, fluid subcooling, and force of acceleration. Secondary parameters such as surface finish and heater material, thickness, and geometry have also been shown to influence the peak heat flux. Empirical correlations are available which account for the formerly mentioned major parameter, but no general correlation which predicts the influence of secondary effects is presently available. In many instances, however, these secondary effects are small, and the magnitude of the peak heat flux may be predicted within engineering accuracy by the following procedure proposed by Gambill (Ref. 12).

The method of Gambill (Ref. 12) is presented in graphical form in Figs. 6, 7, 8, and 9 and Tables V and VI, which contain sufficient information to predict the peak heat flux for saturated and subcooled pool boiling and for subcooled forced convection boiling. The case of bulk or saturated forced convection boiling will be treated in a later report.

The peak heat flux, $(q/A)_{BO}$, is evaluated from the additive formula

$$(q/A)_{BO} = C_F [(q/A)_{sat} - (q/A)_{sub}] + (q/A)_{FC} \tag{15}$$

The terms on the right of Eq. (15) represent contributions to the total peak heat flux for each of the four major phenomena known to influence the magnitude of the heat flux at burnout. Depending on the specific circumstances, only one or more of the terms may be required for a given calculation. The following sections describe means for computing individual terms from the presented figures and tables.

The $(q/A)_{sat}$ term is the peak heat flux obtained in a saturated pool at normal acceleration of gravity when both subcooling and forced convection effects are absent. Its value is calculated from the correlation given by Kutataladze (Ref. 10).

$$\phi(P_r) = (q/A)_{sat} = K h_{fg} \rho_v \left[\frac{\sigma g_c g \Delta \rho}{\rho_v^2} \right]^{\frac{1}{4}} \quad 0.13 < k < 0.17 \tag{16}$$

Since $(q/A)_{sat}$ varies only with pressure, it can be plotted in terms of reduced coordinates as shown in Fig. 6. The vertical axis is ϕ/ϕ_{ref}, where ϕ is determined from Eq. (16) at any arbitrary reduced pressure and ϕ_{ref} is determined from the same equation at a specific reference reduced pressure of 0.05. Reference values, ϕ_{ref}, for a number of fluids have been tabulated in Table V.

Inasmuch as the peak heat flux is an instability-type phenomenon, there is an inherent uncertainty in the absolute value at which it occurs. The ϕ_{max} and ϕ_{min} values tabulated in Table V represent the upper and lower limits of this uncertainty.

The term $(q/A)_{sub}$ appearing in Eq. (15) accounts for the increase in the peak heat flux achieved by subcooling the fluid. Following Zuber (Ref. 13), the subcooling contribution may be determined from

$$\psi(P_r) = \frac{(q/A)_{sub}}{C_\ell \Delta T_{sub}} = K' \sigma g_c g (\Delta \rho)^{0.25} \frac{(\rho \ell)^{0.928}}{(\rho_v)^{0.428}} \tag{17}$$

Again the expression on the right is a function only of pressure, and hence, ψ/ψ_{ref} may be plotted on reduced coordinates in the same manner as described for ϕ/ϕ_{ref}. The result is given in Fig. 7 and Table VI, which are interpreted in direct analogy to Fig. 6 and Table V.

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The factor C_F appearing in Eq. (15) corrects for the departure of the acceleration of gravity, a, from the standard earth's value. Experiments show that in general

$$C_F = \left(\frac{a}{g}\right)^{\frac{1}{4}} \qquad (18)$$

This relationship is plotted in Fig. 8.

Superimposing forced convection on the boiling process results in a further gain in magnitude of the peak heat flux and is accounted for by $(q/A)_{FC}$ in Eq. (15). An expression of the form

$$(q/A)_{FC} = h\Delta T \qquad (19)$$

is used in computing the heat flux contribution by forced convection. The h is the single-phase heat-transfer film coefficient compatible with the flow conditions and geometry of the system. Suitable correlations for h are given in standard references (Refs. 11, 14, and 15). The temperature difference, ΔT, at peak heat flux has been empirically correlated by Bernath (Ref. 16). Figure 9 shows his result. The average fluid velocity, V, added to the temperature difference given by the ordinate of the curve, corrects for thinning of the boiling film with increased velocity.

Having now gained physical insight to the terms comprising Eq. (15), we next consider the detailed procedure followed in numerically evaluating the peak heat flux.

The first step is to write Eq. (15) in the appropriate form for the system being studied.

Possible forms of the equation are:

1. Forced flow, subcooled system

$$(q/A)_{BO} = C_F [(q/A)_{sat\ pool} + (q/A)_{subcooled}] + (q/A)_{FC} \qquad (19a)$$

2. Natural flow, subcooled system

$$(q/A)_{BO} = C_F [(q/A)_{sat\ pool} + (q/A)_{subcooled}] \qquad (19b)$$

3. Natural flow, saturated system

$$(q/A)_{BO} = C_F (q/A)_{sat\ pool} \qquad (19c)$$

A numerical result for $(q/A)_{BO}$ is obtained on substituting values of $(q/A)_{sat\ pool}$, C_F, $(q/A)_{subcooled}$, and $(q/A)_{FC}$ into the suitable

equation. Specific procedures for evaluating these respective quantities are now given.

Evaluation of $(q/A)_{sat}$

1. Evaluate the system design pressure, P,

2. Calculate the reduced pressure, P_r,

3. Find ϕ/ϕ_{ref} from Fig. 6 at the calculated value of P_r,

4. Select $(\phi_{ref})_{max}$ and $(\phi_{ref})_{min}$ for the fluid of interest from Table V,

5. Multiply (ϕ/ϕ_{ref}) separately by $(\phi_{ref})_{max}$ and by $(\phi_{ref})_{min}$ to obtain ϕ_{max} and ϕ_{min}, and

6. Express the saturated pool boiling contribution to the peak heat flux as lying between

$$\phi_{min} < (q/A)_{sat} < \phi_{max}$$

Steps 1 and 6 require further discussion which will be carried out with the view that the system under design is to operate in the nucleate boiling regime (i. e., the local system heat flux must everywhere be less than the minimum peak heat flux). This view is adopted because nucleate boiling is a particularly efficient cooling mechanism, and it is, generally, of most practical interest. Moreover, the destructive burn-out phenomenon which often accompanies the peak heat flux must be carefully guarded against for obvious reasons. However, if it is desired to design for operation in the film boiling regime, one merely reverses the following arguments.

In general, if pressure variations are small, the average system pressure can be employed in step 1. However, a thorough analysis would actually consist of evaluating the pressure distribution and assessing at which location the value, ϕ/ϕ_{ref}, corresponding to the local pressure has a minimum. In the absence of subcooling and forced convection, burnout would occur first at this location, assuming uniform heating over the surface.

With the fluid subcooled and undergoing forced flow, the peak heat flux becomes more sensitive to variations in bulk temperature and velocity than in pressure. Figure 10 illustrates typical trends in $(q/A)_{BO}$ caused by pressure, bulk temperature, and mean velocity variation for the specific case of water flowing in a 1-in.-ID vertical tube. Comparatively speaking, pressure influence is not competitive with that of the other variables when the subcooling is large. Hence,

with significant subcooling and velocity variations in the system, an average pressure may be safely employed in step 1 despite relatively large pressure differences throughout the system.

Directing attention now to step 6, the minimum value of the saturated pool boiling contribution, ϕ_{min}, subject to an appropriate safety factor if burnout is to be rigorously avoided, is the correct design value of the peak heat flux.

The value of $(q/A)_{sat}$ thus determined assumes a gravitational field of $a/g = 1$ with the resulting bouyancy forces directed away from the heated surface. For rotating or swirl flow, a/g is greater than unity, whereas for space applications it is less. Since gravity has direction as well as magnitude, the orientation of the surface with respect to the acceleration vector must also be considered. Account of both these effects is made through the factor, C_F.

Evaluation of C_F

1. Evaluate the magnitude of the acceleration, a, experienced by the boiling surface,

2. Calculate a/g,

3. Find C_F corresponding to the calculated value of a/g from Fig. 8, and.

4. Correct C_F for surface orientation with the multiplicative factor, b, listed in Table VII.

In determining C_F where a varies spatially in either magnitude and/or direction, local values of a and b which minimize C_F are to be used in steps 1 and 4, respectively.

Evaluation of $(q/A)_{sub}$

1. Calculate P_r as in determining $(q/A)_{sat}$,

2. Find ψ/ψ_{ref} corresponding to P_r from Fig. 7,

3. Select $(\psi_{ref})_{min}$ and $(\psi_{ref})_{max}$ for the fluid of interest from Table VI,

4. Calculate ψ_{min} and ψ_{max} by multiplying ψ/ψ_{ref} by $(\psi_{ref})_{min}$ and $(\psi_{ref})_{max}$, respectively,

5. Express $(q/A)_{sub}$ in terms of ψ, C_ℓ, and ΔT_{sub} as

$$(q/A)_{sub} = \psi C_\ell \Delta T_{sub}$$

6. Find the specific heat of the liquid C_ℓ, from tables of property data. The average temperature

$$T_{ref} = (T_{sat} + T_{bulk})/2$$

serves as the correct reference temperature for evaluating C_ℓ,

7. Solve for $(q/A)_{sub}$ by combining C_ℓ, ΔT_{sub}, and ψ as shown in step 5 above, and

8. Express the contribution of subcooling to the peak heat flux as being between

$$[(q/A)_{sub}]_{min} < (q/A)_{sub} < [(q/A)_{sub}]_{max}.$$

Once again it is pointed out that the system under design should be inspected for the location where the contribution of $(q/A)_{sub}$ to the overall peak heat flux is such that $(q/A)_{total}$ has a minimum value, if burnout is to be avoided.

Determination of (q/A) Forced Convection

1. Calculate the single-phase forced convection heat-transfer coefficient, h, from the appropriate correlation given in standard references for the system geometry of interest. For example, the value of h for flow through a cylindrical duct may be calculated from the Dittus-Boelter correlation

$$h = 0.023 \frac{k_{bulk}}{D} (N_{RE})_{bulk}^{0.8} (N_{PR})_{bulk}^{0.4}$$

The fluid properties are evaluated at T_{ref}.

2. Find from Fig. 9 the value of $(T_{w_{BO}} - T_{sat} + 0.45 \, V)$ corresponding to the reduced pressure of the system,

3. Compute the average fluid flow velocity in units of ft/sec (eg, $V = \frac{\dot{m}}{\rho A}$),

4. Calculate $(T_w - T_{sat})_{BO}$ from the results of steps 2 and 3 (eg, $(T_w - T_{sat})_{BO} = $ (Value read from curve) $- 0.45 \, V$),

5. Determine the overall temperature difference from

$$\Delta T = (\Delta T_{sat})_{BO} + \Delta T_{sub}$$

6. Find the forced convection contribution to the burnout heat flux from

$$(q/A)_{FC} = h\Delta T$$

The foregoing procedure outlined for determining the peak nucleate boiling heat flux is now illustrated with a number of example calculations.

Example 1: Saturated water at atmospheric conditions is boiling on the horizontal base of a flat-bottomed container. Find the peak heat flux for the system.

Since this is a pool boiling problem with no subcooling, Eq. (15) reduces to

$$(q/A)_{BO} = C_F (q/A)_{sat}$$

The heater surface is horizontal, and the local acceleration of gravity is not altered, hence $a/g = 1.0$, and C_F is found from Fig. 8 and Table VII to be equal to 1.0. The remaining term, $(q/A)_{sat}$, can be found from the ratio, ϕ/ϕ_{ref}, selected from Fig. 6 at the reduced system pressure

$$P_r = 14.7/3206 = 0.0046$$

Figure 6 gives

$$\phi/\phi_{ref} = 0.405$$

From Table V ϕ_{min} and ϕ_{max} for water are 850,500 and 1,112,000, respectively.

Thus,

$$(\phi/\phi_{ref})\phi_{min} < (q/A)_{sat} < (\phi/\phi_{ref})\phi_{max}$$
$$344,000 < (q/A)_{sat} < 450,000$$

The above result expresses the fact that burnout could be expected to occur at a heat flux of as low as 317,000 Btu/hr-ft^2; consequently, a safe design criterion should be based on this minimum value multiplied by an appropriate safety factor. On the other hand, the maximum value 450,000 Btu/hr-ft^2 denotes the upper limit which, because of the statistical nature of the burnout phenomena, might possibly be obtained before the burnout crises take place. This value is useful in the design of a system required to operate under film boiling conditions. That is, above 450,000 Btu/hr-ft^2, one may be reasonably confident that the boiling process has undergone the transition from nucleate to film boiling.

Example 2: Evaluate the contribution to the peak heat flux from the various terms appearing in Eq. (15) for water flowing

in a vertical 1-in.-ID tube at a pressure of 50 psia, a subcooling of 100°F, and a velocity of 10 ft/sec.

For this problem Eq. (15) is

$$(q/A)_{BO} = C_F \left[(q/A)_{sat} - (q/A)_{sub} \right] + (q/A)_{FC}$$

The reduced pressure of water corresponding to 50 psia is 0.0156. From Fig. 6 the value of ϕ/ϕ_{ref} is found to be 0.67. Confining attention to the minimum value of the burnout heat flux, $(\dot{\phi}_{ref})_{min}$ from Table V, is 850,500 Btu/hr-ft^2, and on multiplication the saturated pool boiling contribution becomes

$$(q/A)_{sat} = (\phi/\phi_{ref})(\phi_{ref})_{min} = (0.67)(850,500)$$
$$= 570,000 \text{ Btu/hr-ft}^2$$

The subcooling contribution is found from

$$(q/A)_{sub} = (\psi_{ref})_{min}(\psi/\psi_{ref}) C_\ell \Delta T_{sub}$$

Referring to Fig. 7, ψ/ψ_{ref} at a reduced pressure of 0.0156 has a value of 1.76. Table VI gives $(\psi_{ref})_{min}$ for water as 4280 lb$_m$/hr-ft^2, hence,

$$(q/A)_{sub} = (4280)(1.76) C_\ell \Delta T_{sub}$$
$$= 7530 \, C_\ell \, \Delta T_{sub}$$

Recall that C_ℓ is to be evaluated at a reference temperature given by

$$T_{ref} = \frac{T_{sat} + T_{bulk}}{2}$$

By adding and subtracting T_{sat} from the right-hand side, this may be written

$$T_{ref} = T_{sat} - \frac{\Delta T_{sub}}{2}$$

The saturation temperature of water at 50 psia is 281°F, hence,

$$T_{ref} = 281 - \frac{100}{2} = 231°F$$

The specific heat of water from property tables is 1.0 Btu/lb$_m$-°F, and the contribution of subcooling to the peak heat flux is determined as:

$$(q/A)_{sub} = \left(7530 \, \frac{lb_m}{hr\text{-}ft^2} \right) \left(1.0 \, \frac{Btu}{lb_m\text{-}°F} \right) (100°F)$$
$$= 753,000 \text{ Btu/hr-ft}^2$$

The forced convection effect $(q/A)_{FC}$ is computed from

$$(q/A)_{FC} = h\Delta T$$

where the single-phase heat-transfer coefficient for the tube geometry can be calculated from

$$h = 0.023 \frac{k_{bulk}}{D} (N_{RE})^{0.8}_{bulk} (N_{PR})^{0.4}_{bulk}$$

and ΔT can be determined from Fig. 9. Evaluation of the well-known correlation for the film coefficient gives h = 3059 Btu/hr-ft^2-°F where the fluid properties are based on bulk temperature

$$T_{bulk} = T_{sat} - \Delta T_{sub}$$
$$= 181°F$$

At P_r = 0.0156, ΔT_{BO} is (Fig. 9) 77°F, and ΔT is

$$\Delta T = \Delta T_{BO} + \Delta T_{sub} + 0.45 V$$

$$= 77 + 100 + (0.45)(10 \frac{ft}{sec}) °F$$

$$= 182°F$$

Thus

$$(q/A)_{FC} = (3059 \text{ Btu/hr-ft}^2\text{-°F})(182°F)$$

$$= 556,000 \text{ Btu/hr-ft}^2$$

Finally, the total peak heat flux is found by substitution of the preceding results into Eq. (15) where a value of 0.75 is assigned to C_F (see Table VII) to account for the vertical orientation of the heated surface.

$$(q/A)_{BO} = [0.75[570,000 + 753,000] + 556,000] \text{ Btu/hr-ft}^2$$

$$= 1,550,000 \text{ Btu/hr-ft}^2$$

The present prediction technique was compared with 1134 data points for water and nonaqueous fluids by Gambill (Ref. 12). The maximum deviation of 95 percent of this data was 46 percent. All of the data were within 82 percent. The maximum deviation of the calculated values of ϕ and ψ for the fluids, water, oxygen, hydrogen, nitrogen, and neon, from the best-fit line obtained with the correspondence principle was 12 and 15 percent, respectively.

SECTION V
FILM BOILING

Nucleate boiling (Section III) undergoes a transition to film boiling when the boiling heat flux exceeds the peak value described in Section IV. Interestingly, however, film boiling thus established does not revert back to nucleate boiling by simply reducing the heat flux once again below the peak value. In fact, a return to nucleate boiling is only achieved when the heat flux is substantially decreased and reaches a value referred to as the minimum critical heat flux (Section VI). The path followed during the transition from nucleate boiling to film boiling and back is illustrated in Fig. 11.

The name film boiling is descriptive of this form of boiling since the surface on which the boiling occurs is literally covered with a film of vapor. The insulating effects of the vapor film greatly restrict heat transfer from the surface, and hence, film boiling does not constitute an efficient mechanism for cooling a given system. Despite this fact, however, many industrial heat exchangers or refrigerators employing volatile refrigerants or liquid cryogenics, having low critical heat flux values, must, of necessity, operate with film boiling taking place in the tube bundles or transfer lines. Thus, design correlations for quantitative predictions of the heat transfer associated with film boiling are needed.

Stable film boiling has been analyzed mathematically by several investigators who assume a model wherein heat is transferred by conduction and radiation across the vapor film. In general, however, the analytical results have required empirical corrections when compared with available experimental data. Moreover, the equations differ with orientation of the heated surface, and consequently, separate equations are required for vertical and horizontal surfaces. The following paragraphs present a method of calculating the heat transfer for a vertical or horizontal surface using respectively different equations expressed in terms of common dimensionless factors.

The total film boiling heat-transfer coefficient is given by a convective contribution, h_c, and a radiation contribution, h_r, as shown below

$$h = h_c + (3/4) h_r \tag{20}$$

Breen and Westwater (Ref. 17) have found that the convective heat-transfer coefficient, h_c, for film boiling on horizontal tubular surfaces can be predicted from the equation

$$h_c (\lambda_c)^{1/4} / F = 0.59 + 0.069 \, \lambda_c / D \tag{21}$$

where

$$F = \left(\frac{g \, K_v^3 \rho_v (\rho_\ell - \rho_v) L'}{\mu_v \, \Delta T_{sat}} \right)^{\frac{1}{4}}$$

$$\lambda_c = 2\pi (\sigma / (\rho_\ell - \rho_v))^{\frac{1}{2}}$$

$$L' = h_{fg} [1 + 0.34 \, C_v \, \Delta T_{sat} / h_{fg}]^2$$

Notice that for a flat plate, D approaches infinity, and the last term of Eq. (21) vanishes.

For a vertical surface, Seader et al (Ref. 18) proposes the following equation

$$\frac{h_c (L)^{\frac{1}{4}}}{F} = 0.80 \tag{22}$$

where L is the length of the vertical surface.

The radiation contribution to Eq. (20) is found from

$$h_r = \frac{\sigma'(T_w^4 - T_{sat}^4)}{\left(\frac{1}{\epsilon_w} + \frac{1}{a\ell} - 1 \right) \left(T_w - T_{sat} \right)}$$

ϵ_w = emissivity of the solid surface on which the boiling takes place

α_ℓ = absorptivity of the liquid (generally unity)

σ' = Stefan-Boltzmann constant

No consideration of subcooling or forced flow effects is included in Eqs. (21) and (22). A discussion of these will be given in a later report.

To simplify the calculation of h_c for any given fluid, F and λ_c have been plotted versus reduced pressure, P_r, in Figs. 12 and 13, respectively. On the vertical scale of Fig. 12, F = $F(P_r, \Delta T_{sat})$ has been normalized with the value of F = F(0.05, ΔT_{sat}), for which case the ratio F/F_{ref} for all fluids lies approximately on a common curve with ΔT_{sat} as a parameter. The reduced pressure, P_r = 0.05, at which F_{ref} is evaluated is selected arbitrarily. Values of F_{ref} for a number of fluids have been tabulated in Table VIII. Figure 13 shows $(\lambda_c)/(\lambda_c)_{ref}$ plotted in an analogous manner to that of F/F_{ref}; in this case, however, the ΔT_{sat} dependence is so small it has been omitted. Values of $(\lambda_c)_{ref}$ are also tabulated in Table VIII.

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In Figs. 14 and 15, $h_r\left(\dfrac{1}{\epsilon_w} + \dfrac{1}{\alpha_\ell} - 1\right)$ has been plotted versus ΔT_{sat} with T_w as a parameter. The absorptivity of the liquid, α_ℓ, in general will have a value of unity.

The steps required to calculate h for film boiling are summarized as follows:

1. Determine the design pressure of the system.

2. Calculate the reduced pressure, P_r.

3. From Figs. 12 and 13 find F/F_{ref} and $(\lambda_c)/(\lambda_c)_{ref}$ corresponding to the calculated value of P_r. $\left[(\lambda_c)/(\lambda_c)_{ref}\right.$ need not be determined for a vertical surface.$\Big]$

4. From Table VIII select the values of F_{ref} and $(\lambda_c)_{ref}$ for the fluid of interest. $\left[(\lambda_c)_{ref}\right.$ need not be determined for a vertical surface.$\Big]$

5. Calculate F and λ_c employing the expressions

$$F = (F/F_{ref})\, F_{ref}$$

and

$$\lambda_c = (\lambda_c/(\lambda_c)_{ref})\, (\lambda_c)_{ref}$$

respectively.

6. Find h_c for a horizontal surface from

$$h_c = 0.59 + 0.069\ \lambda_c/D\ (F/\lambda_c^{1/4})$$

or for a vertical surface from

$$h_c = 0.80\ F/(L)^{1/4}$$

Note λ_c and L raised to the one-fourth power can be quickly evaluated from Fig. 16.

7. Find h_r from either Fig. 14 or 15.

8. Calculate the total heat-transfer coefficient from

$$h = h_c + 0.75\, h_r$$

9. Determine the heat flux using the relationship

$$q/A = h\Delta T_{sat}$$

In the above description of the stepwise procedure, ΔT_{sat} was assumed known, and q/A was to be computed. In many practical problems q/A will be known, and it will be necessary to determine the unknown wall temperature, T_w. This can be done by a trial and error procedure which converges rapidly because of the small effect of ΔT_{sat} on F/F_{ref}. This small dependence of F/F_{ref} on ΔT_{sat} is apparent from Fig. 12.

Further inspection of Fig. 12 also illustrates that the film boiling heat flux will increase with increasing pressure for the vertical surface, inasmuch as h is directly proportional to F. This is, in turn, true for horizontal surfaces since the magnitude of λ_c/D is small compared with $F/\lambda_c^{1/4}$. For systems wherein the pressure varies, the selection of the appropriate system pressure made in step 1 should be in accordance with the expected influence of pressure on q/A.

Application of the foregoing method is now illustrated with the following example problem.

Example 1: A pool of saturated liquid nitrogen at 14.7 psia surrounds a long 0.5-in. horizontal stainless steel rod which is at a constant temperature of 600°R. Determine the heat flux from the rod.

$$q/A = h \Delta T$$

where

$$\Delta T = T_w - T_{sat}$$

$$= 600 - 139°R$$

$$= 461°R$$

The heat-transfer coefficient, h, is given by

$$h = h_c + 0.75 h_r$$

where h_c is evaluated from Eq. (21). The value of F is found by entering Fig. 12 at a reduced pressure of 0.0298 and at a ΔT of 461°F. The F/F_{ref} is approximately 0.88. From Table VIII, F_{ref} for nitrogen at $\Delta T = 461°R$ is approximately 16.0; therefore,

$$F = (F/F_{ref}) F_{ref}$$

$$= (0.88) (16.0)$$

$$= 14.1 \frac{Btu}{hr\text{-}ft^{7/4}\text{-}°F}$$

Analogously

$$\lambda_c = (\lambda_c/(\lambda_c)_{ref}) \, (\lambda_c)_{ref}$$

$$= (1.05) \, (0.021)$$

$$= 0.022 \text{ ft}$$

From Fig. 14 $\lambda_c^{1/4} = 0.39$ ft $^{1/4}$.

Thus

$$h_c = \left[0.59 + 0.069 \, \frac{(0.022 \text{ ft})}{(0.0417 \text{ ft})} \right] \left[\frac{\dfrac{Btu}{14.1 \text{ hr-ft}^{7/4}\text{-}^\circ F}}{0.39 \text{ ft}^{1/4}} \right] \cdot$$

$$h_c = 22.7 \, \frac{Btu}{\text{hr-ft}^2\text{-}^\circ R}$$

The radiation heat-transfer coefficient is taken from Fig. 14 at $\Delta T = 416^\circ R$

$$h_r \left[\frac{1}{\epsilon_w} + \frac{1}{\alpha_\ell} - 1 \right] = 0.52$$

Let $\epsilon_w = 0.6$ and $\alpha_\ell = 1$.

Then $h_r = 0.31$

$$h = 22.9 \, \frac{Btu}{\text{hr-ft}^2\text{-}^\circ R}$$

and

$$(q/A) = \left(23 \, \frac{Btu}{\text{hr-ft}^2\text{-}^\circ R} \right) (416)^\circ R$$

$$q/A = 9568 \, \frac{Btu}{\text{hr-ft}^2\text{-}^\circ R}$$

It is now necessary to check if the system is actually in the film boiling regime or if it is still in the nucleate boiling regime. To do this, the peak heat flux is determined from Section IV, and the corresponding ΔT_{sat} is found from Section III. If ΔT_{sat} is greater than the above value, the system is undergoing film boiling.

The peak heat flux at $P_r = 0.0298$ is determined by finding $\phi/\phi_{ref} = 0.85$ from Fig. 6 and $(\phi_{ref})_{min} = 59,000$ and $(\phi_{ref})_{max} = 77,200$ from Table V. Thus the peak heat flux lies between 50,200 and 65,600 Btu/hr-ft^2.

30

· From Section III, $K/K_{ref} = 1.2$ at $P_r = 0.0298$ and $K_{ref} = 0.913°R/(Btu/hr-ft^2)^{0.3}$. The value of ΔT_{sat} is then

$$(\Delta T_{sat})_{min} = (K/K_{ref}) \, (K_{ref}) \, (q/A)_{min}^{0.3}$$

$$= (1.2) \, (0.913) \, (25)$$

$$= 27°R$$

$$(\Delta T_{sat})_{max} = (1.2) \, (0.913) \, (28)$$

$$= 31°R$$

The system is in film boiling, since $\Delta T_{sat} = 461°R$ is much greater than $(\Delta T_{sat})_{max} = 31°R$.

A comparison of the accuracy of the proposed method of computation is illustrated in Fig. 17. The outlined area represents the region within which a representative sample of the available experimental data lies. The dashed lines represent the maximum deviation of Eq. (21) calculated for seven typical fluids from a curve through the averaged values. Thus, the area enclosed by the dashed lines is indicative of the error associated with representing the ratio F/F_{ref} for several fluids with a single curve, that is, the error involved in employing the correspondence principle. The accuracy of the correspondence principle is obviously sufficient relative to the experimental uncertainty.

SECTION VI
MINIMUM FILM BOILING HEAT FLUX

The transition from film boiling to nucleate boiling occurs at a heat flux known as the minimum film boiling heat flux, $(q/A)_{MC}$. Most theories used to predict the magnitude of this heat flux are based on hydrodynamic instability theories. Agreement of the theory with experiment has been good. However, because of the fact that the phenomenon occurs as a result of an instability, there is an inherent uncertainty in the value of $(q/A)_{MC}$. Thus, only the range within which $(q/A)_{MC}$ lies can be determined.

Prediction of the minimum heat flux is normally of lesser importance than prediction of the maximum or peak heat flux in that the transition to nucleate boiling is generally accompanied with a marked reduction in surface temperature and consequently, no severe thermal loading of the system as described in Section IV. One does, however, encounter a

need to predict the minimum heat flux in transient cooling problems where a high temperature surface is suddenly cooled by a volatile coolant. In such a situation, the surface normally experiences film boiling initially with a change to nucleate boiling when a critical surface temperature is reached. The several orders of magnitude change in the heat-transfer coefficient accompanying the boiling transition becomes a design consideration in many instances.

Zuber (Ref. 13) derived the following correlation to predict the minimum heat flux

$$(q/A)_{MC} = const\, (\rho_v)_{film}\, h_{fg}\, \frac{g\sigma(\rho_\ell - \rho_v)^{\frac{1}{4}}}{(\rho_\ell + \rho_v)^2} \qquad (26)$$

where all properties are based on the saturation temperature except $(\rho_v)_{film}$, which is evaluated at the average vapor film temperature. The constant has a value between $0.09 > const > 0.18$ which reflects the inherent uncertainty associated with the instability phenomenon.

Transposing $(\rho_v)_{film}$ results in the right-hand side being a function solely of system pressure.

$$\Omega = \frac{(q/A)_{MC}}{(\rho_v)_{film}} = f(P) \qquad (27)$$

Thus, Ω may be plotted as a function of reduced pressure. Further, normalizing Ω with an arbitrarily chosen reference value, Ω_{ref} (in this case, Ω_{ref} is determined from Eq. (27) by evaluating it at $P_r = 0.05$), Ω/Ω_{ref} may be plotted as a single curve (Fig. 18) which is applicable, within the accuracy of the correlation, to most all fluids.

In Table IX values of Ω_{ref} for 16 different fluids have been tabulated. This table coupled with Fig. 18 permits rapid evaluation of $(q/A)_{MC}$ through the following outlined procedure:

1. Determine the operating pressure of the system,

2. Compute the reduced pressure, P_r,

3. From Fig. 18 select Ω/Ω_{ref} corresponding to P_r calculated in step 2,

4. Find Ω_{ref} for the fluid of interest from Table IX, and

5. Compute the range of the minimum heat flux from
   $(\rho_v)_{film}\, (\Omega/\Omega_{ref})\, (\Omega_{ref})_{min} < (q/A)_{MC} < (\rho_v)_{film}\, (\Omega/\Omega_{ref})$
   $(\Omega_{ref})_{max}$

It is now necessary to describe the procedure for determining $(\rho_v)_{film}$. The density of the vapor film, $(\rho_v)_{film}$, is to be determined from property tables for the fluid of interest at a film temperature equal to $(T_w + T_{sat})/2$. However, inasmuch as $(q/A)_{MC}$ is not a function of T_w, and in general, T_w is not known a priori, a trial and error procedure must be used to find the correct value of $(\rho_v)_{film}$. The suggested procedure is as follows:

1. Initially determine $(\rho_v)_{film}$ based on the saturation temperature, T_{sat},

2. From the result of step 5 above, compute $(q/A)_{MC}$,

3. With this value of $(q/A)_{MC}$, use the film boiling correlation described in Section V to determine T,

4. Reevaluate $(\rho_v)_{film}$ based on $T_{film} = (T_w + T_{sat})/2$, and

5. If necessary, repeat steps 1 through 4 above until the procedure converges.

An illustrative example calculation is now presented to illuminate the details of the foregoing prediction technique.

> Example: A long, heated 1-ft-diam rod is dropped into a large pool of liquid nitrogen maintained at atmospheric pressure. Film boiling immediately occurs on the rod. As the rod cools, find the minimum critical heat flux and the surface temperature at which the transition to nucleate boiling takes place. The reduced pressure of liquid nitrogen is

$$P_r = 0.0299$$

From Fig. 18

$$\Omega/\Omega_{ref} = 1.05$$

and from Table IX

$$(\Omega_{ref})_{max} = 3009 \ \frac{Btu\text{-}lb_m}{hr\text{-}ft^5}$$

and

$$(\Omega_{ref})_{min} = 2310 \ \frac{Btu\text{-}lb_m}{hr\text{-}ft^5}$$

Thus

$$(\rho_v)_{film} (2200) (1.05) < (q/A)_{MC} < (\rho_v)_{film} (4330) (1.05)$$

$$2430 \ (\rho_v)_{film} < (q/A)_{MC} < 3160 \ (\rho_v)_{film} \ \frac{Btu}{hr\text{-}ft^2}$$

To determine $(\rho_v)_{film}$, initially assume $(\rho_v)_{sat} = 0.28 \dfrac{lbm}{ft^3}$.

Then

$$680 \text{ Btu/hr-ft}^2 < (q/A)_{MC} < 885 \text{ Btu/hr-ft}^2$$

The temperature difference for film boiling at this heat flux is found from Section V as follows:

At $P_r = 0.0299$, Figs. 12 and 13 give $F/F_{ref} = 0.45$; $\lambda/(\lambda_c)_{ref} = 1.05$, respectively.

From Table VIII

$$F_{ref} = 16.9 \, (\lambda_c)_{ref} = 0.021$$

In evaluating the above parameters, ΔT on the order of $100°R$ has been assumed. This value will be checked in the final calculation.

The influence of radiation on the film boiling heat-transfer coefficient near the minimum critical value is generally insignificant and hence will be neglected here.

Thus

$$F = (0.45)\,(16.9), \quad \lambda_c = (1.05)\,0.021$$

$$F = 7.61 \, \frac{Btu}{hr\text{-}ft^{\frac{1}{4}}\text{-}°F}, \lambda_c = 0.022 \text{ ft}$$

and

$$h_c = \left[0.59 + 0.069\left(\frac{0.022 \text{ ft}}{1.0 \text{ ft}}\right)\right] \frac{7.61}{0.39} \, \frac{Btu}{hr\text{-}ft^2\text{-}°F}$$

$$h_c = 11.5 \, \frac{Btu}{hr\text{-}ft^2\text{-}°F}$$

The temperature difference ΔT is given by $\Delta T = \dfrac{q/A}{h}$ which in our case becomes

$$\frac{680}{11.5} \,(°R) < \Delta T < \frac{885}{11.5} \,(°R)$$

$$59°R < \Delta T < 77°R$$

The wall temperature is found from

$$T_w = \Delta T + T_{sat}$$

hence,

$$203°R < T_w < 221°R$$

The film temperature is given by

$$T_{film} = \frac{T_w + T_{sat}}{2}$$

therefore

$$174°R < T_{film} < 183°R$$

Adjusting the value of ρ, to correspond with $T_{film} = 178°R$ gives

$$(\rho_v)_{film} = 0.21 \frac{lb_m}{ft^3}$$

and

$$510 \frac{Btu}{hr\text{-}ft^2} < (q/A)_{MC} < 663 \frac{Btu}{hr\text{-}ft^2}$$

The value of ΔT becomes

$$\frac{510}{11.5}°R < \Delta T < \frac{663}{11.5}°R$$

$$44°R < \Delta T < 58°R$$

Repeating the iteration once more gives

$$522 \frac{Btu}{hr\text{-}ft^2} < (q/A)_{MC} < 680 \frac{Btu}{hr\text{-}ft^2}$$

and

$$45°R < \Delta T < 60°R$$

Since ΔT lies between the above limits, the assumption $\Delta T = 100$ made in calculating h_c is sufficiently accurate.

The final result may be summarized as follows. The rod will under-go a transition from film boiling to nucleate boiling at a heat flux between 680 and 522 Btu/hr-ft². The wall temperature at that time will be approximately 197°R (average of maximum and minimum values).

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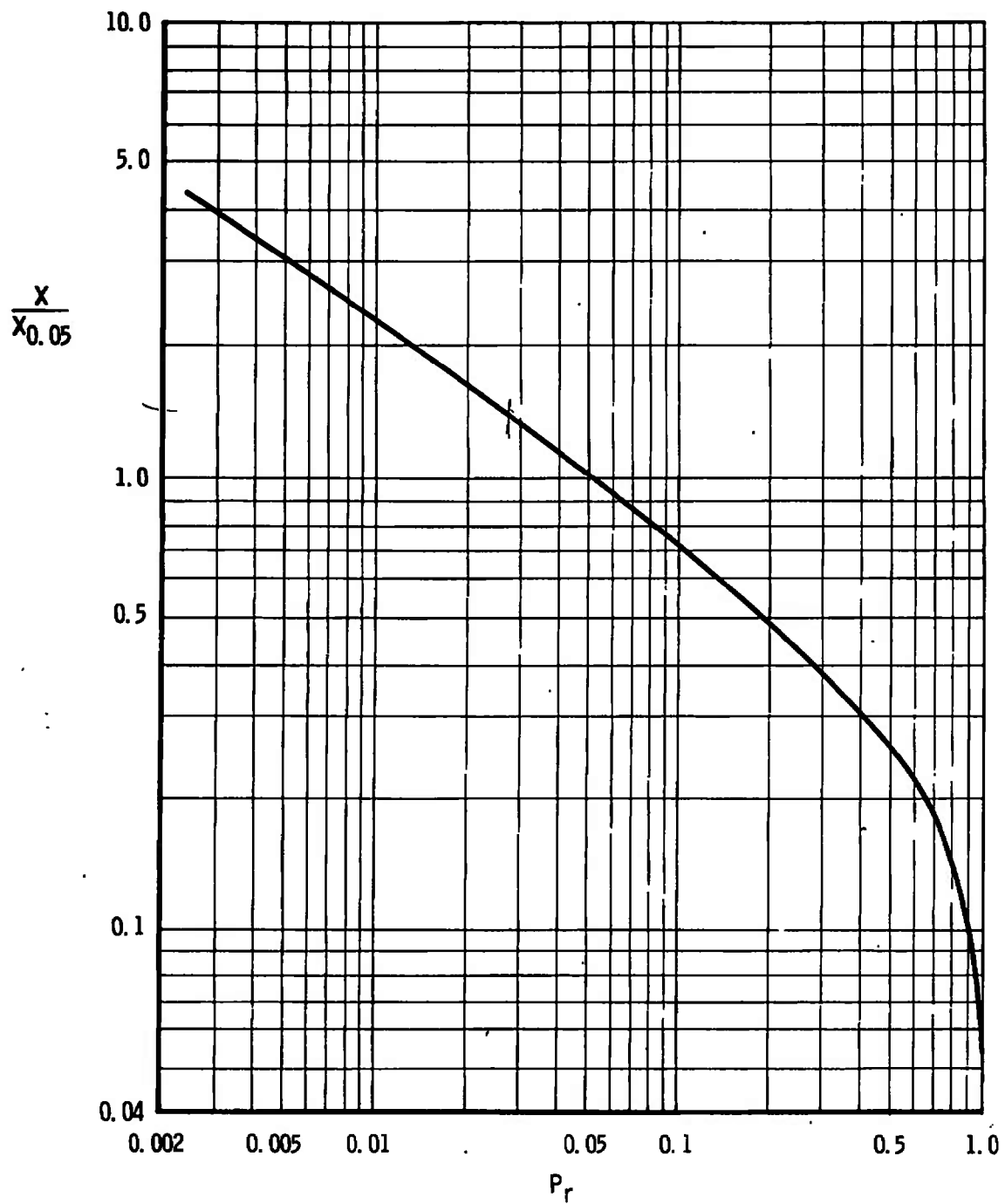APPENDIXES
I. ILLUSTRATIONS
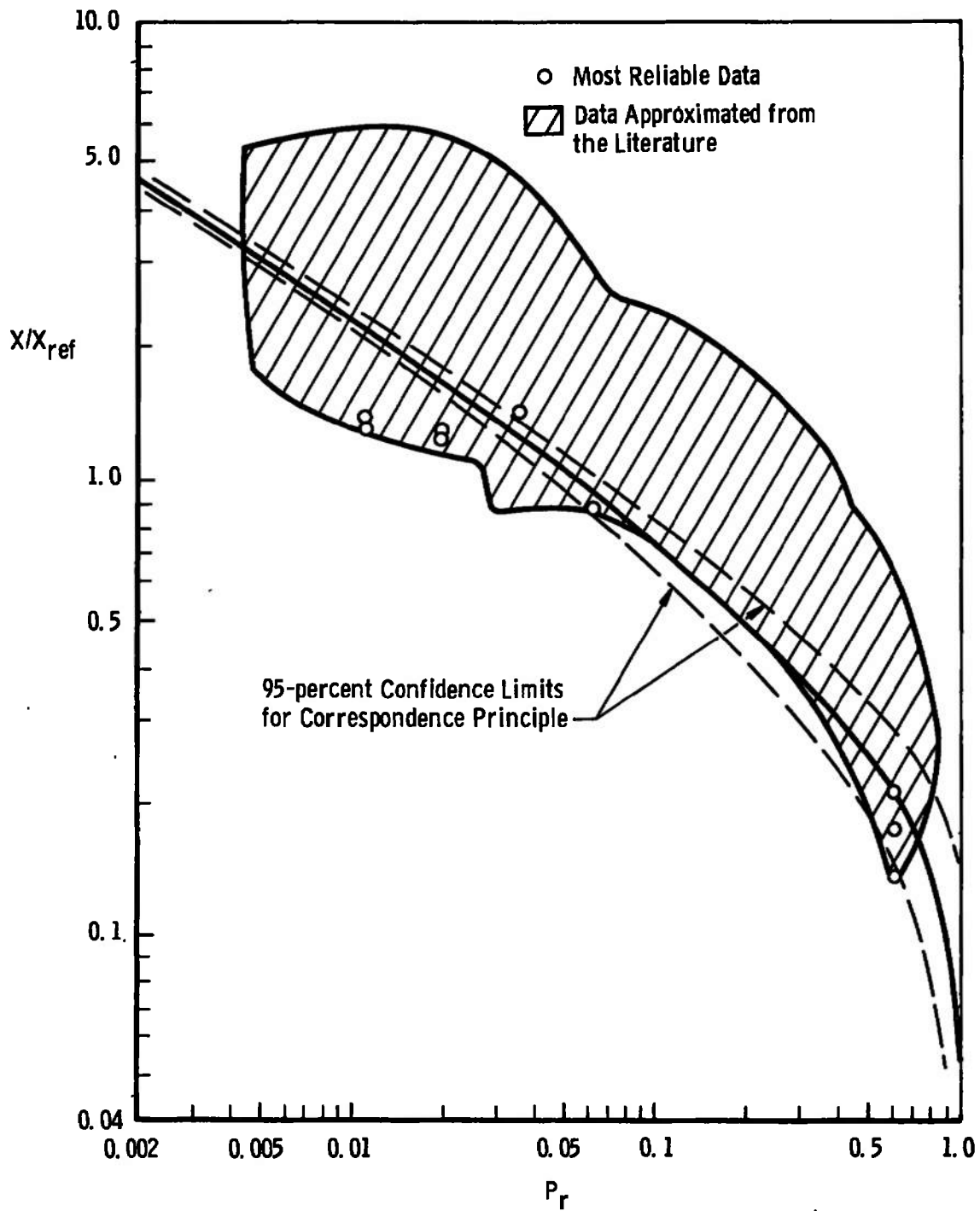II. TABLES

Fig. 1 Incipient Boiling Parameter

Fig. 2 Comparison of the Range of Experimental Data on Incipient Boiling with Accuracy of the Method Based on the Correspondence Principle
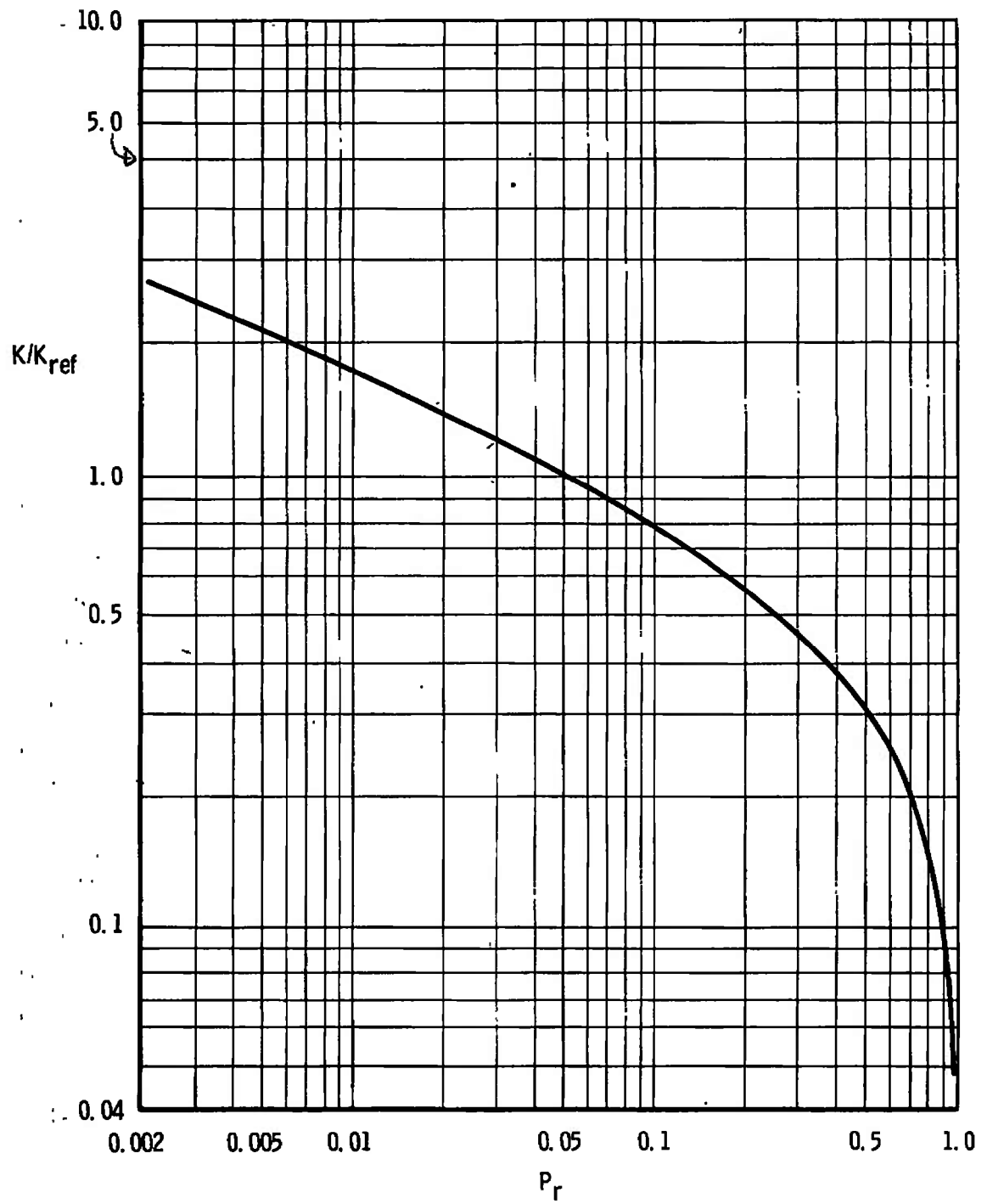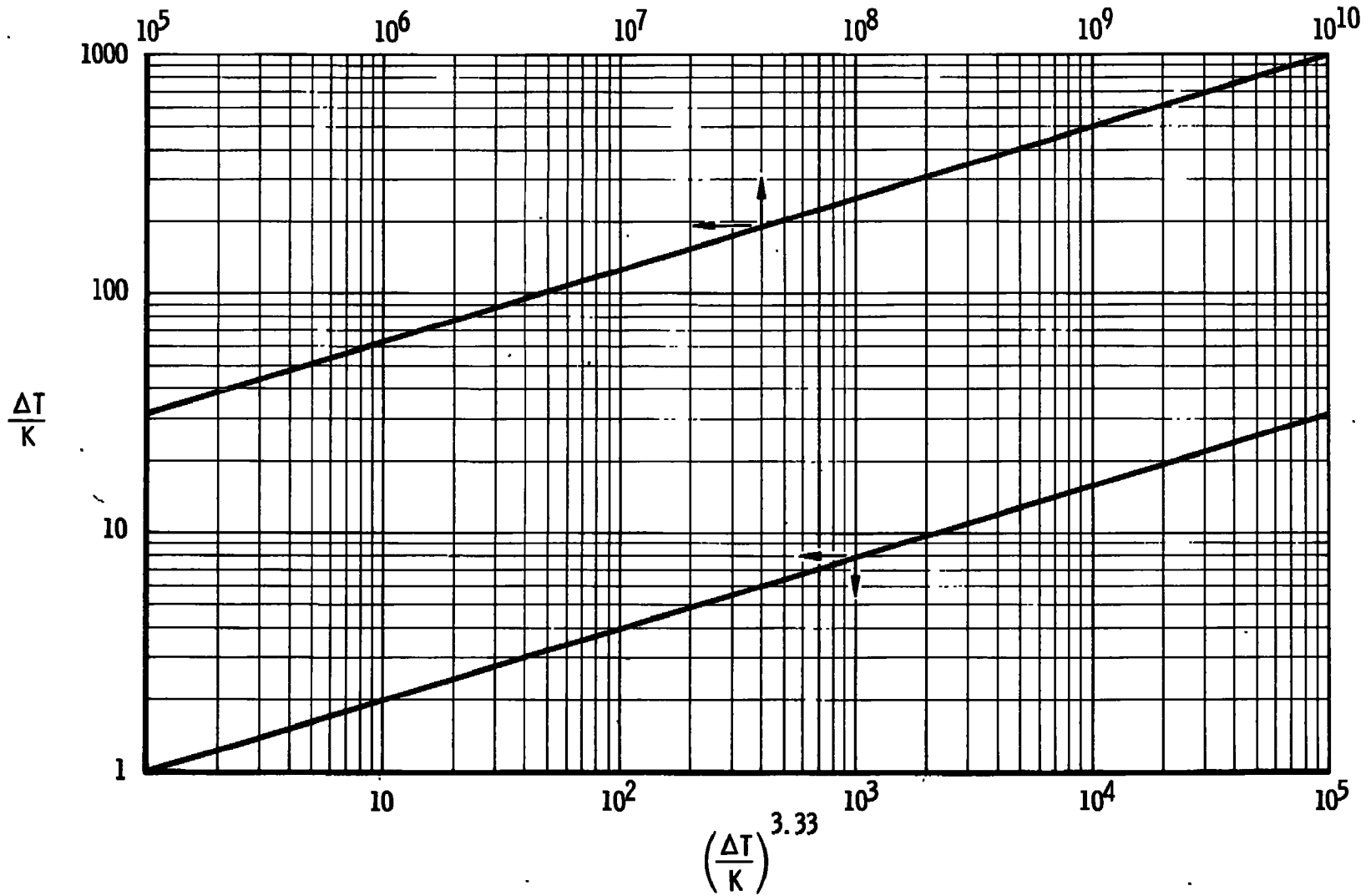
Fig. 3 Nucleate Boiling Parameter

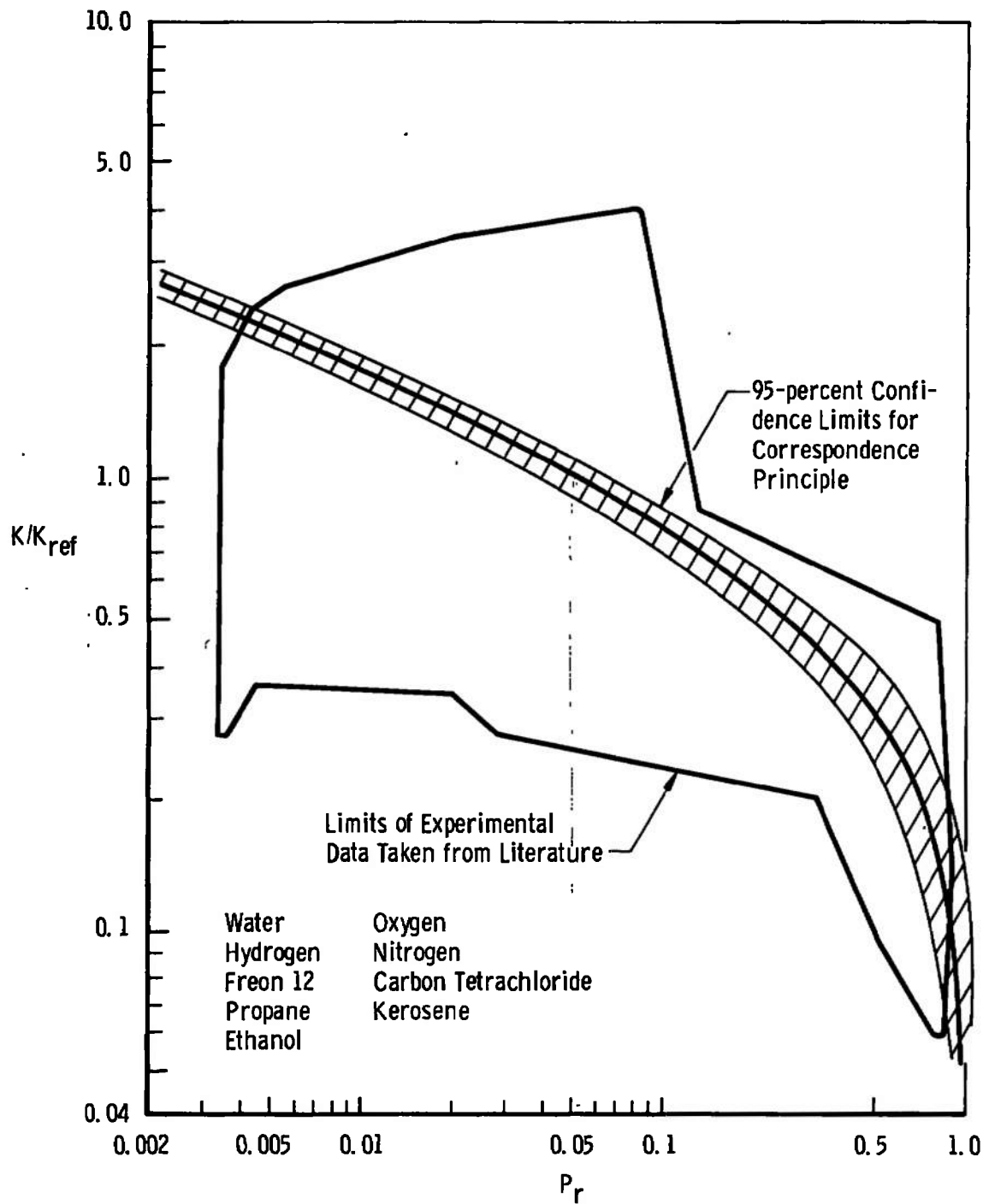Fig. 4 Plot of (ΔT/L) Raised to the 3.33 Power

Fig. 5 Comparison of the Range of Experimental Data on Nucleate Boiling with Accuracy of the Method Based on the Correspondence Principle
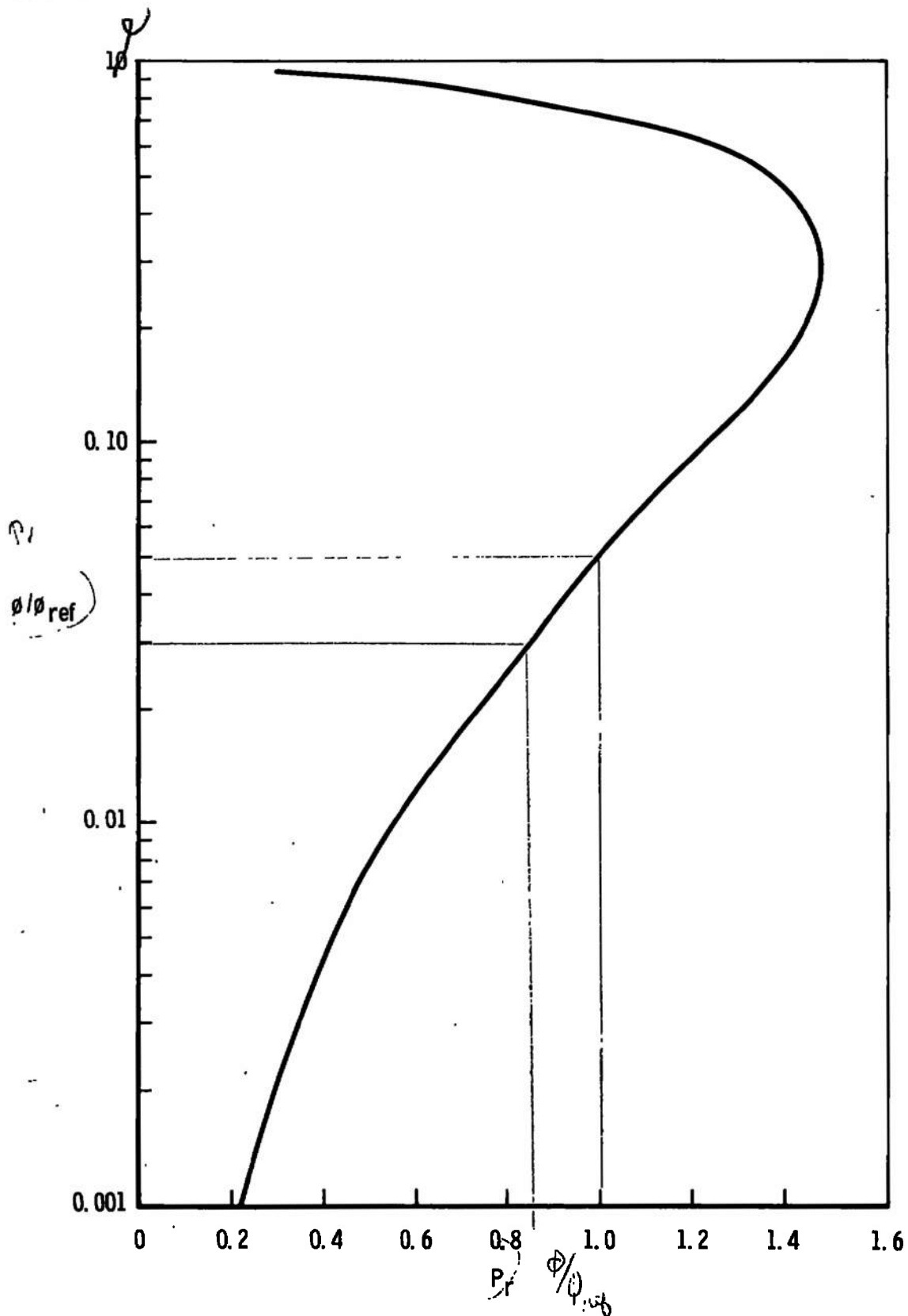
Fig. 6 Saturated Pool Boiling Burnout Heat Flux Parameter
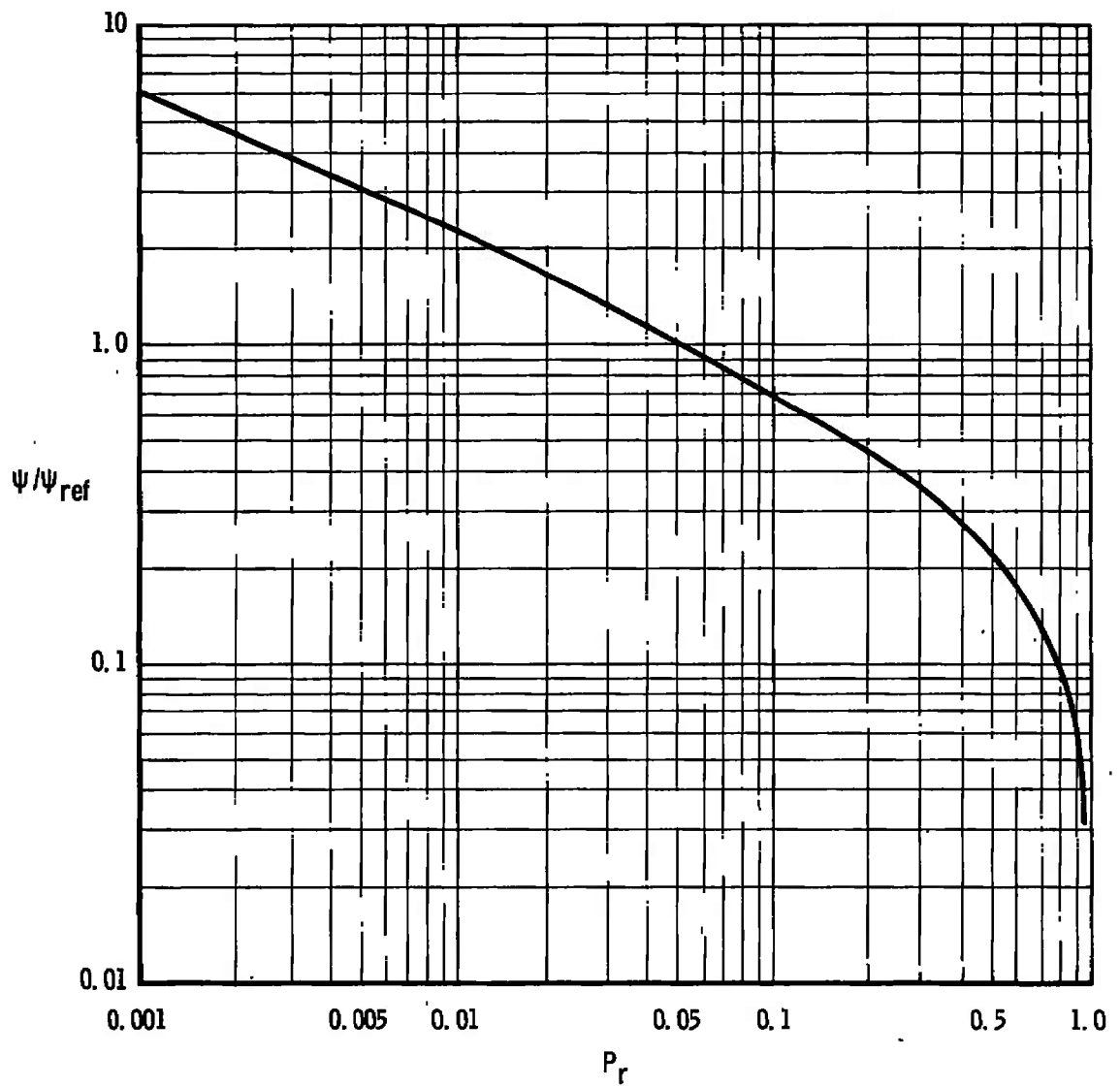
Fig. 7 Burnout Heat Flux Subcooling Parameter
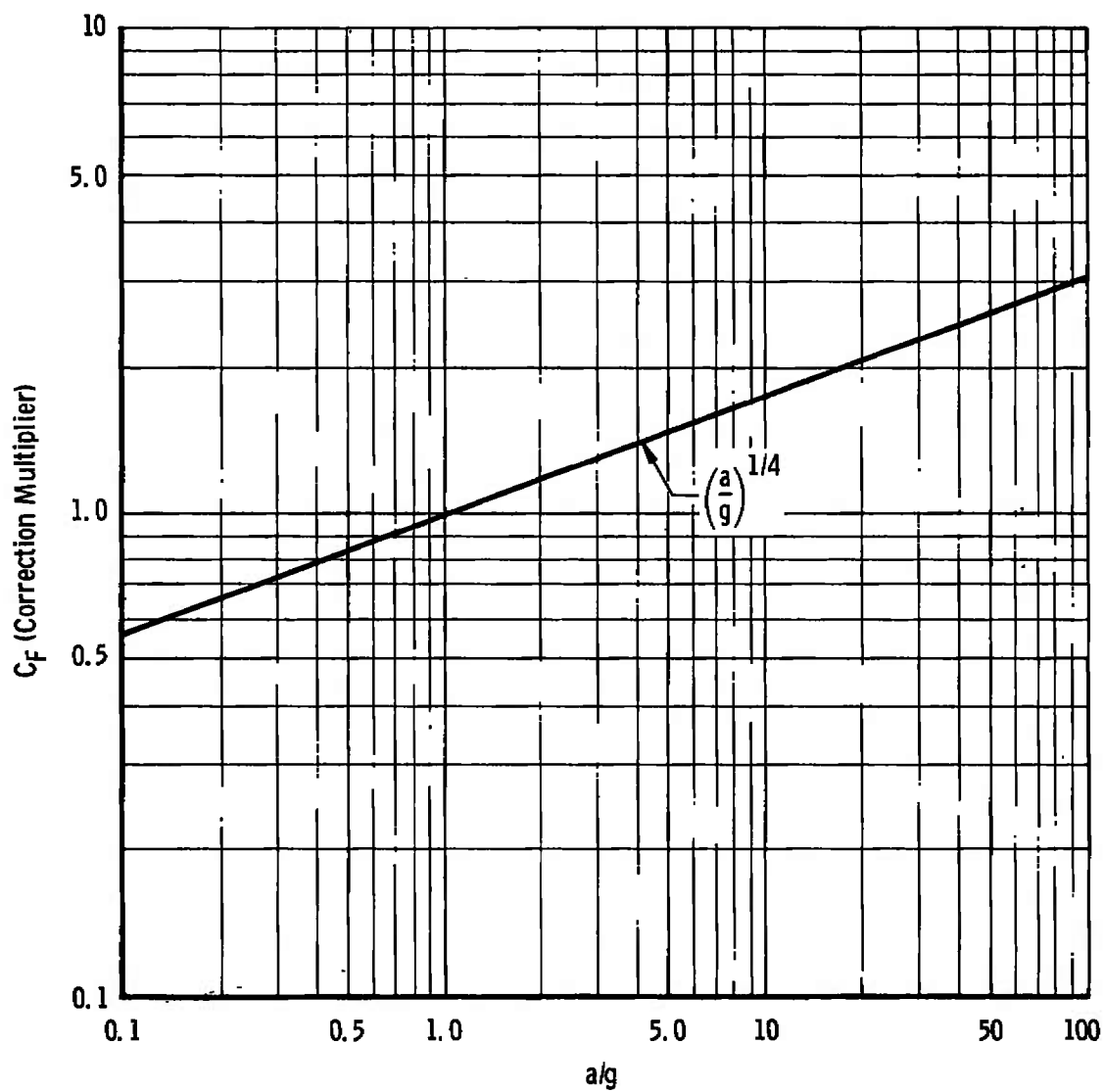
Fig. 8 Factor for Correcting Burnout Heat Flux for the Influence of Changes in Local Acceleration
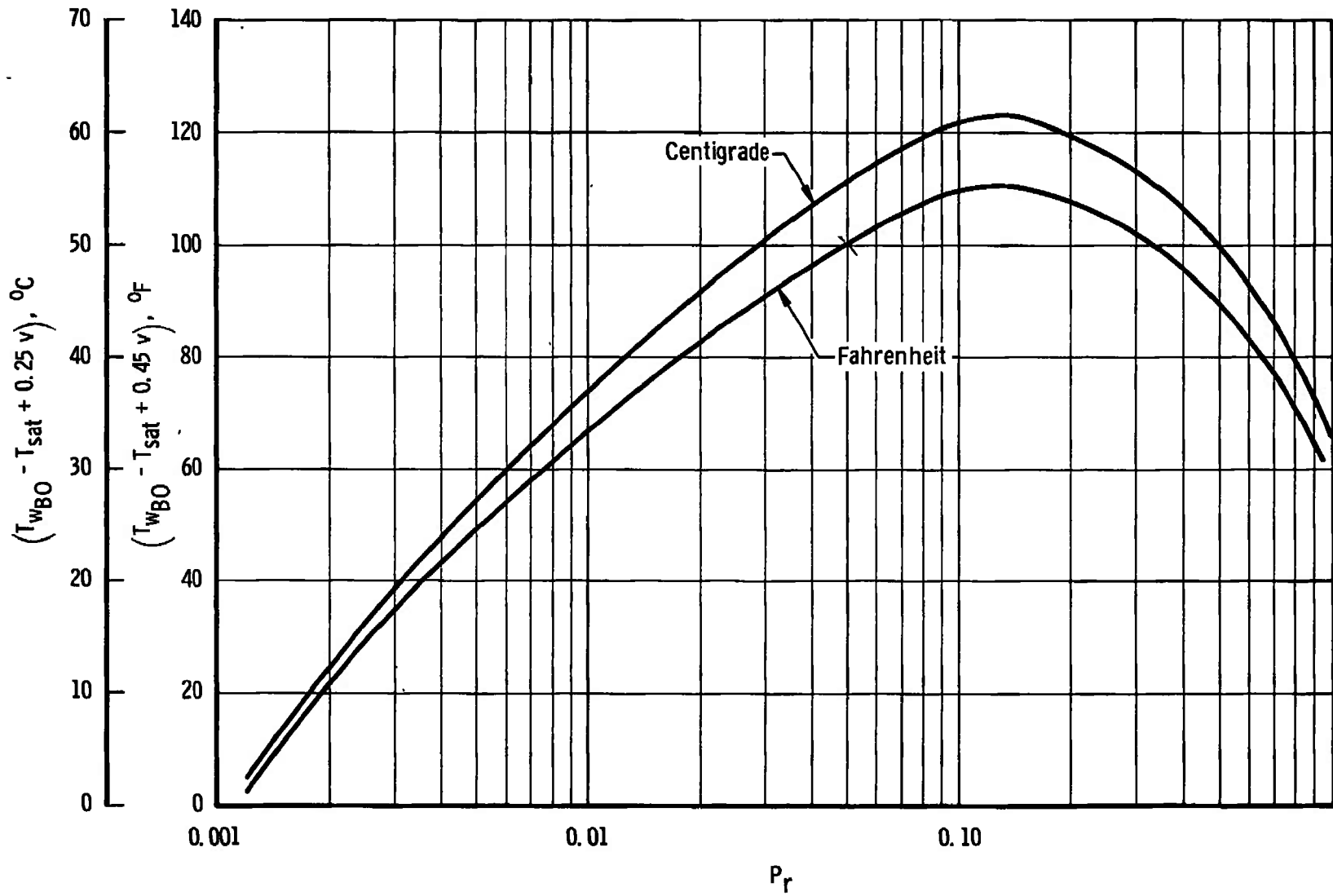
Fig. 9 Wall Temperature at the Burnout Heat Flux
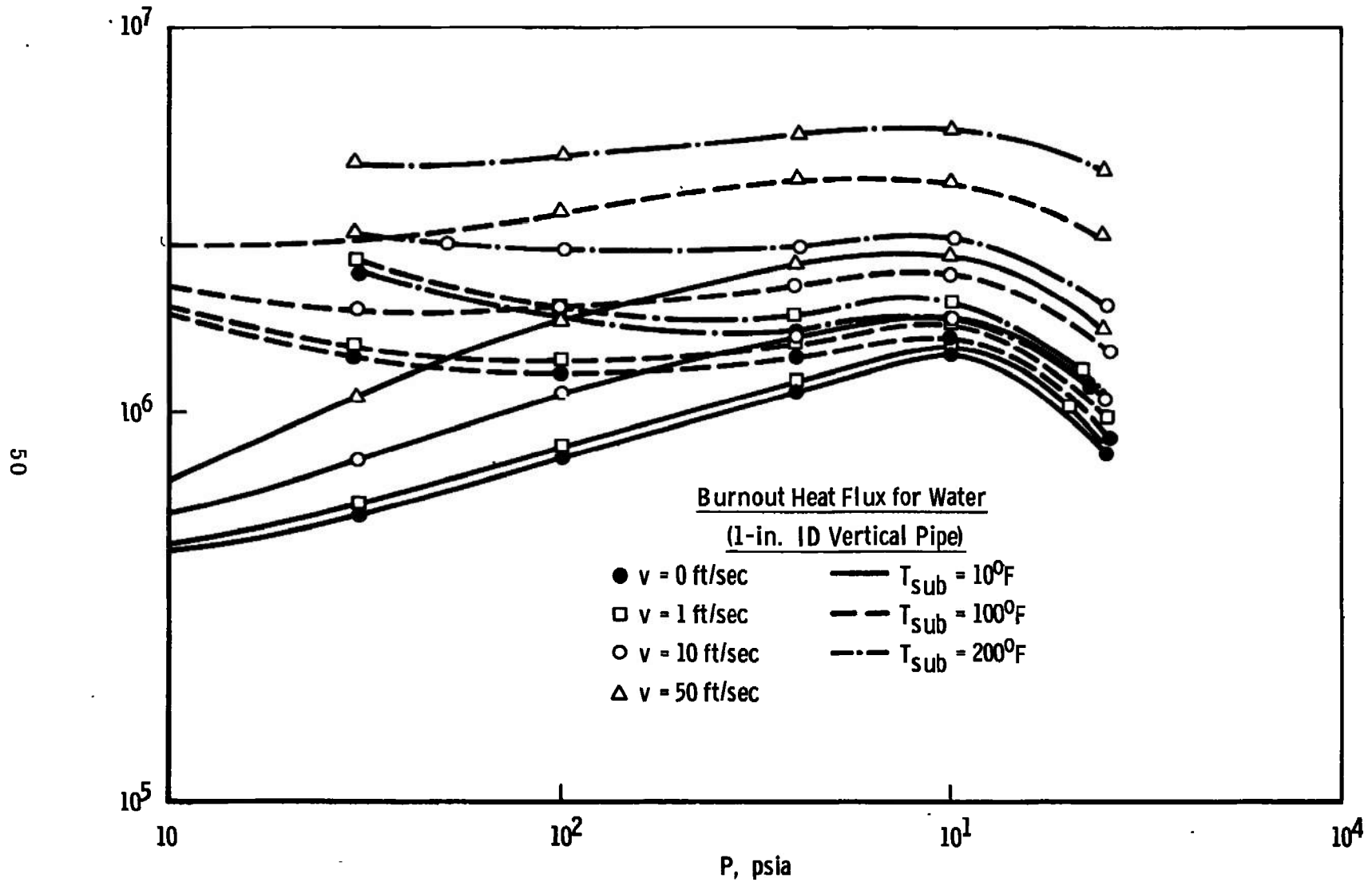
Fig. 10 Influence of the Principal Variables on Burnout Heat Flux in Water
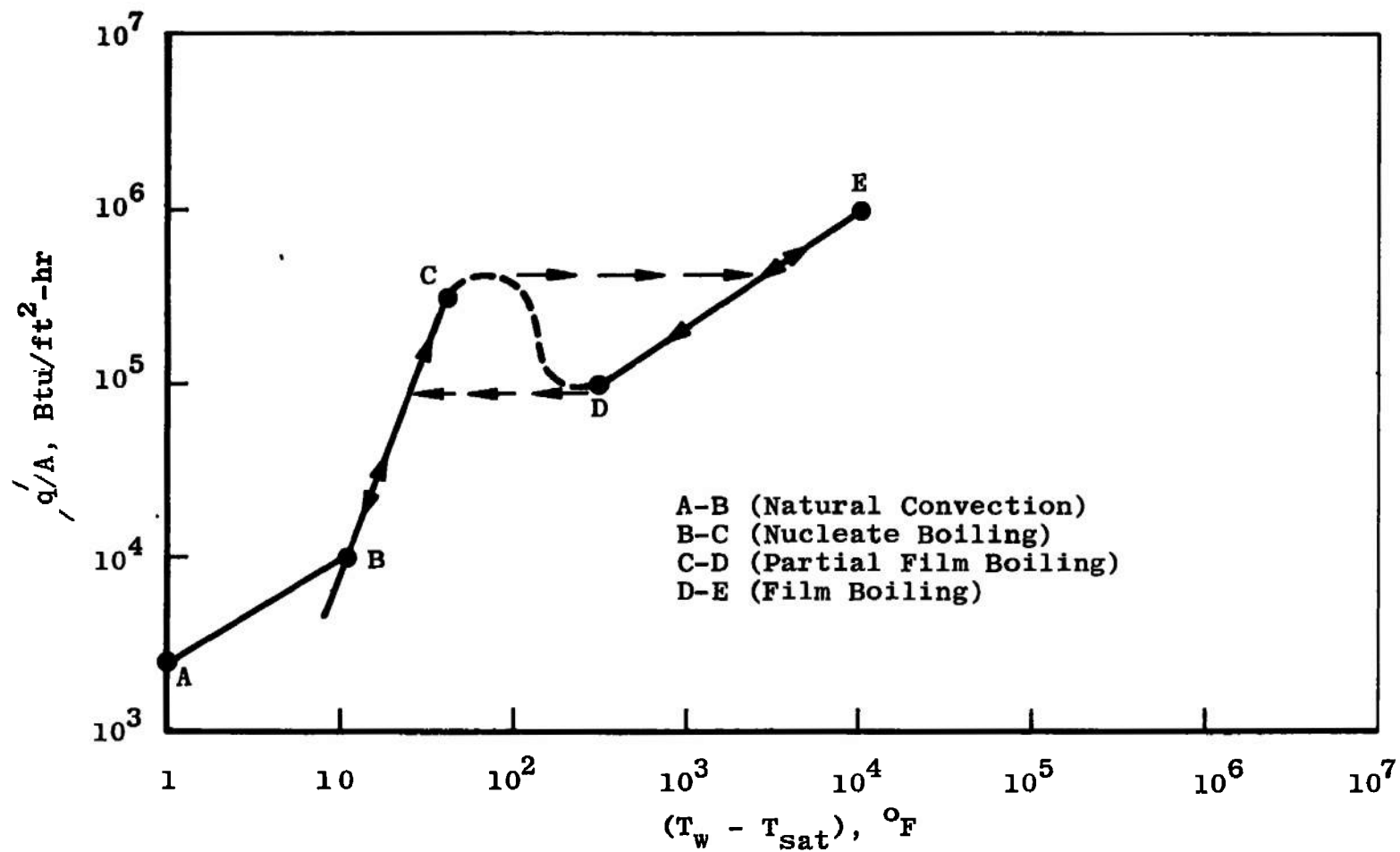
Fig. 11 Boiling Regimes Encountered in Pool Boiling
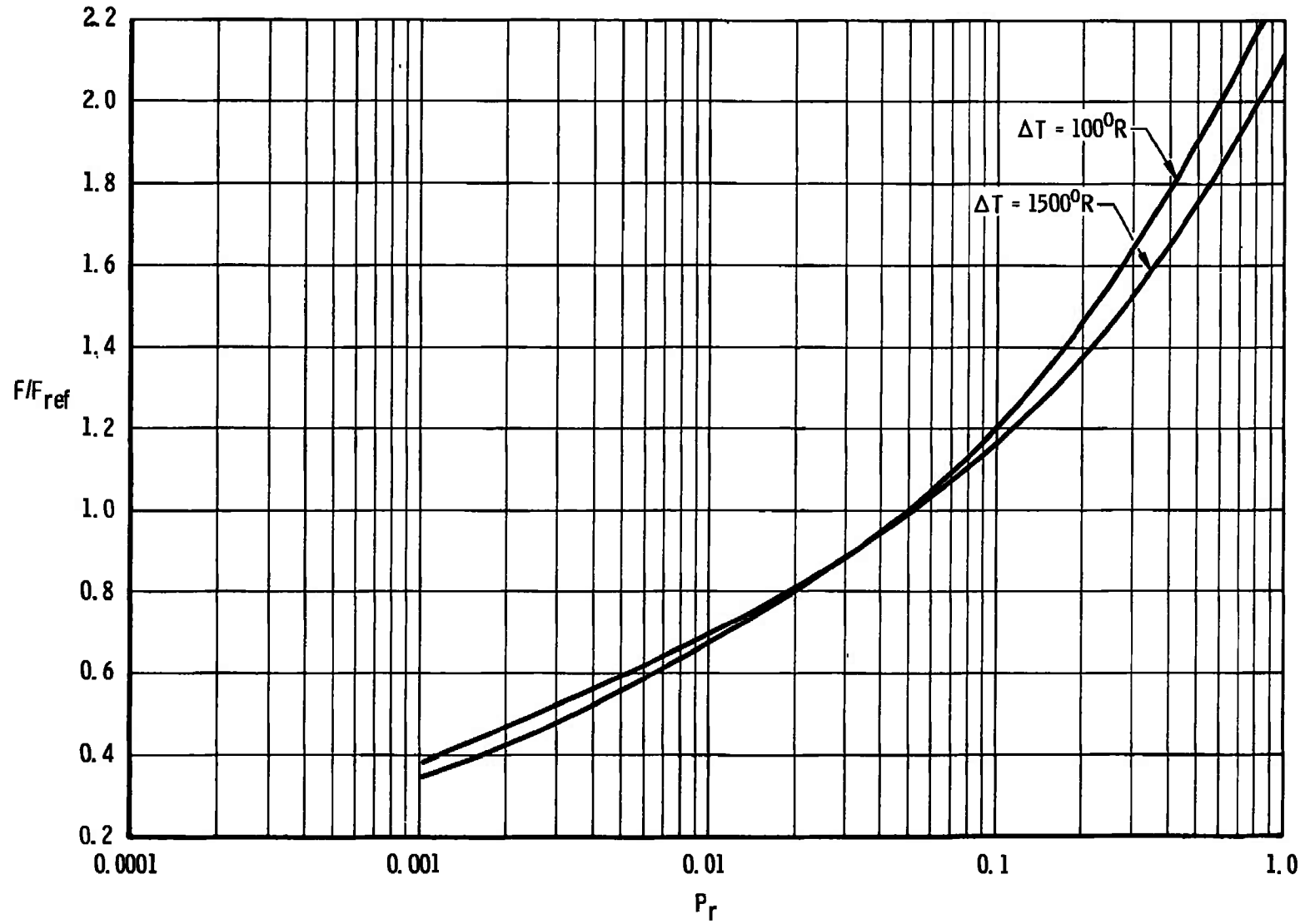
Fig. 12 Critical Wavelength Parameter Used in Film Boiling Calculation
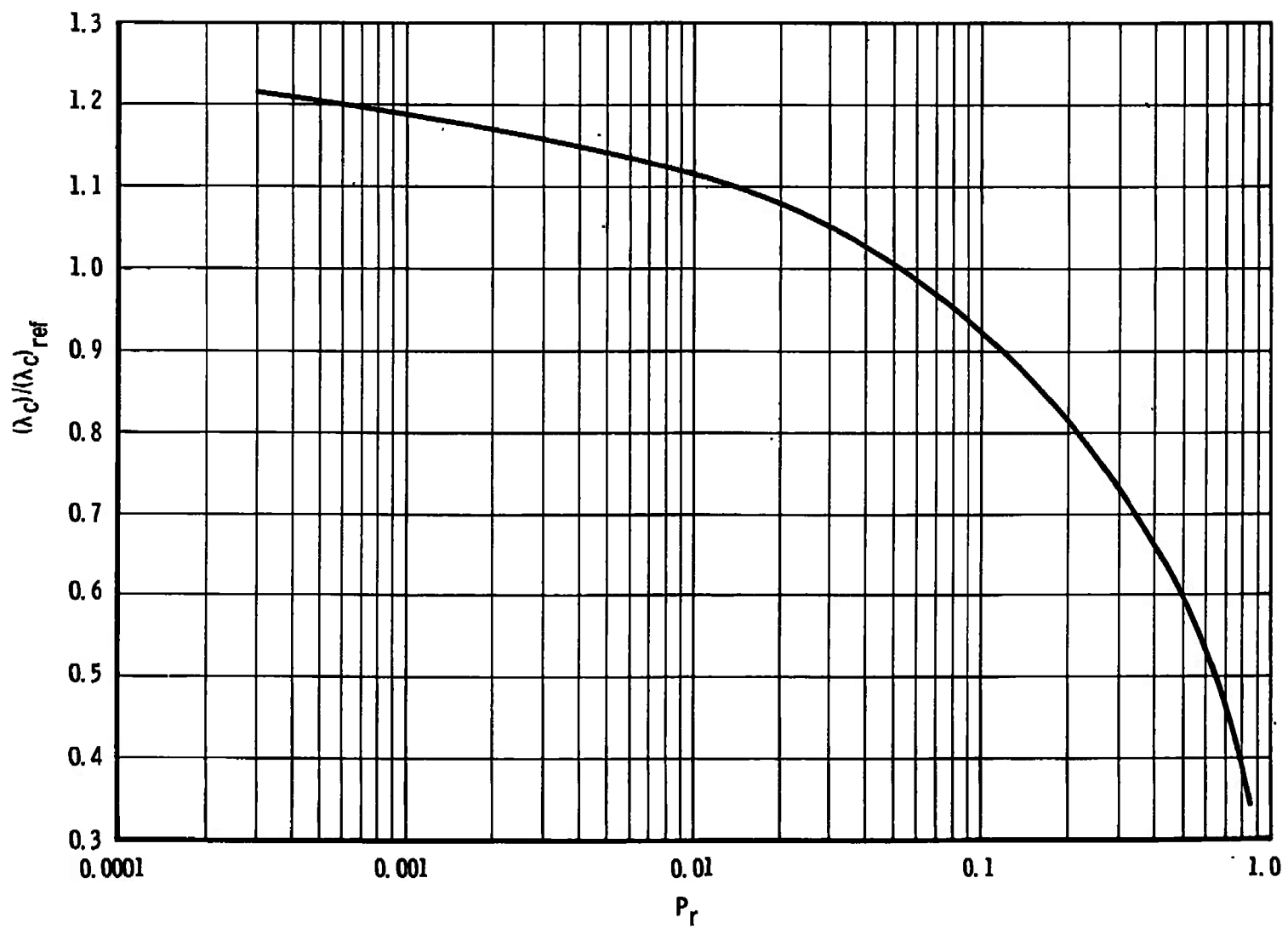
Fig. 13 Critical Wavelength Parameter Used in Film Boiling Calculation

$$h_r \left[\frac{1}{\varepsilon_w} + \frac{1}{\alpha_\ell} - 1 \right] = \frac{\sigma \left[T_w^4 - T_{sat}^4 \right]}{\Delta T_{sat}}$$
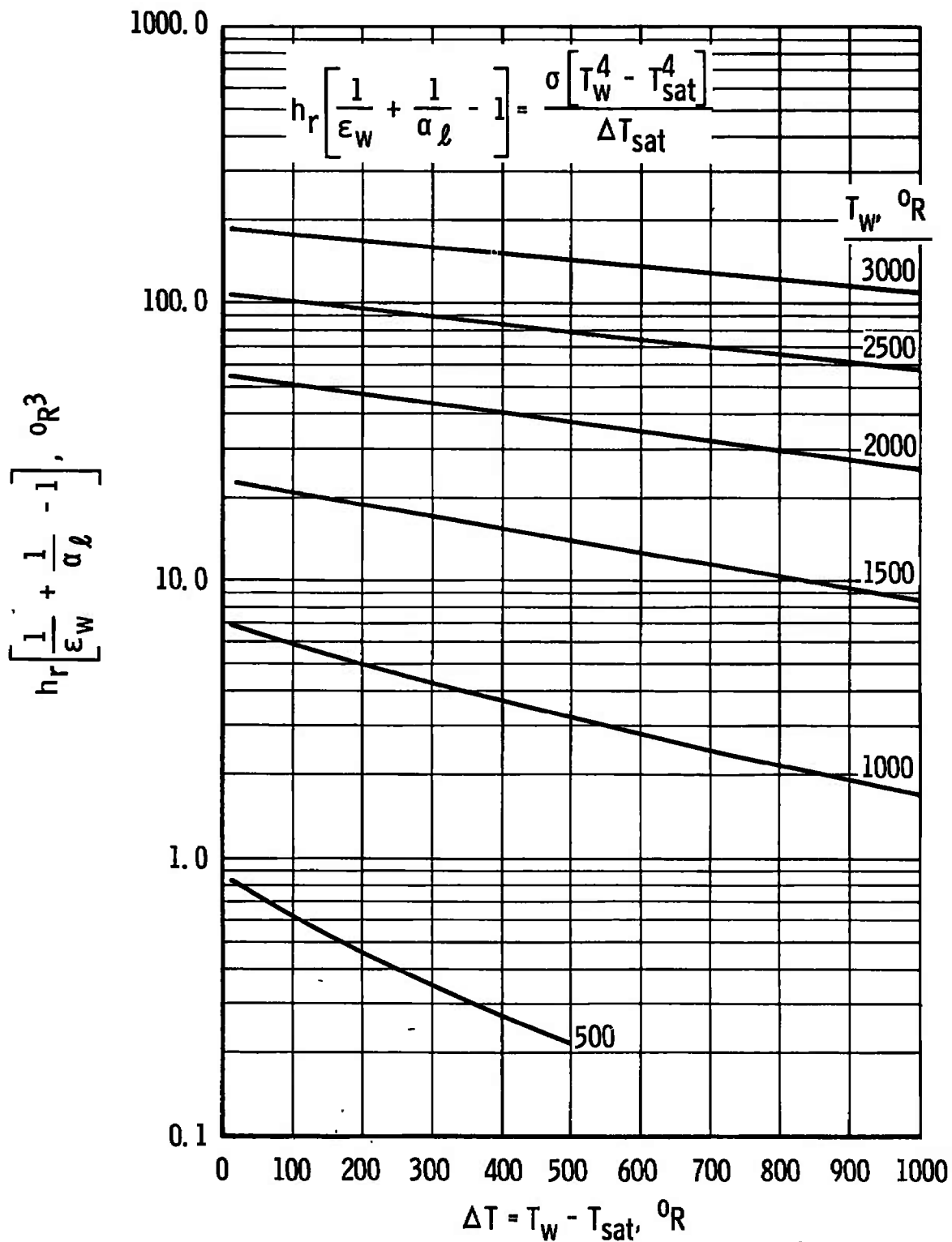
Fig. 14 Additive Radiation Heat-Transfer Contribution to the Overall
Film Boiling Heat-Transfer Coefficient (Range from 500 to 3000°R)

$$h_r \left[\frac{1}{\varepsilon_w} + \frac{1}{a_\ell} - 1 \right] = \frac{\sigma \left(T_w^3 - T_c^4 \right)}{\Delta T_{sat}}$$
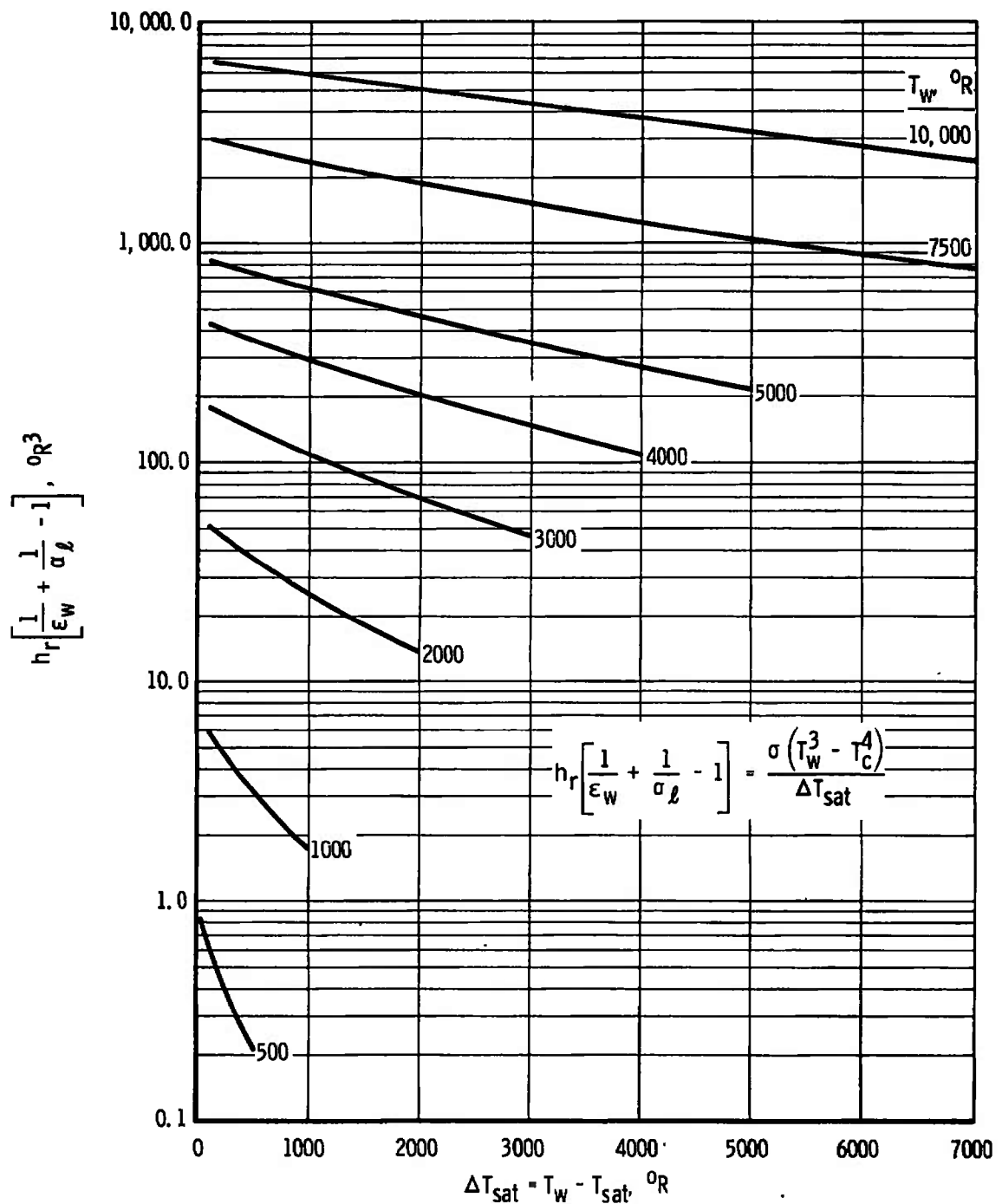
Fig. 15 Additive Radiation Heat-Transfer Contribution to the Overall Film Boiling Heat-Transfer Coefficient (Range from 500 to 10,000°R)
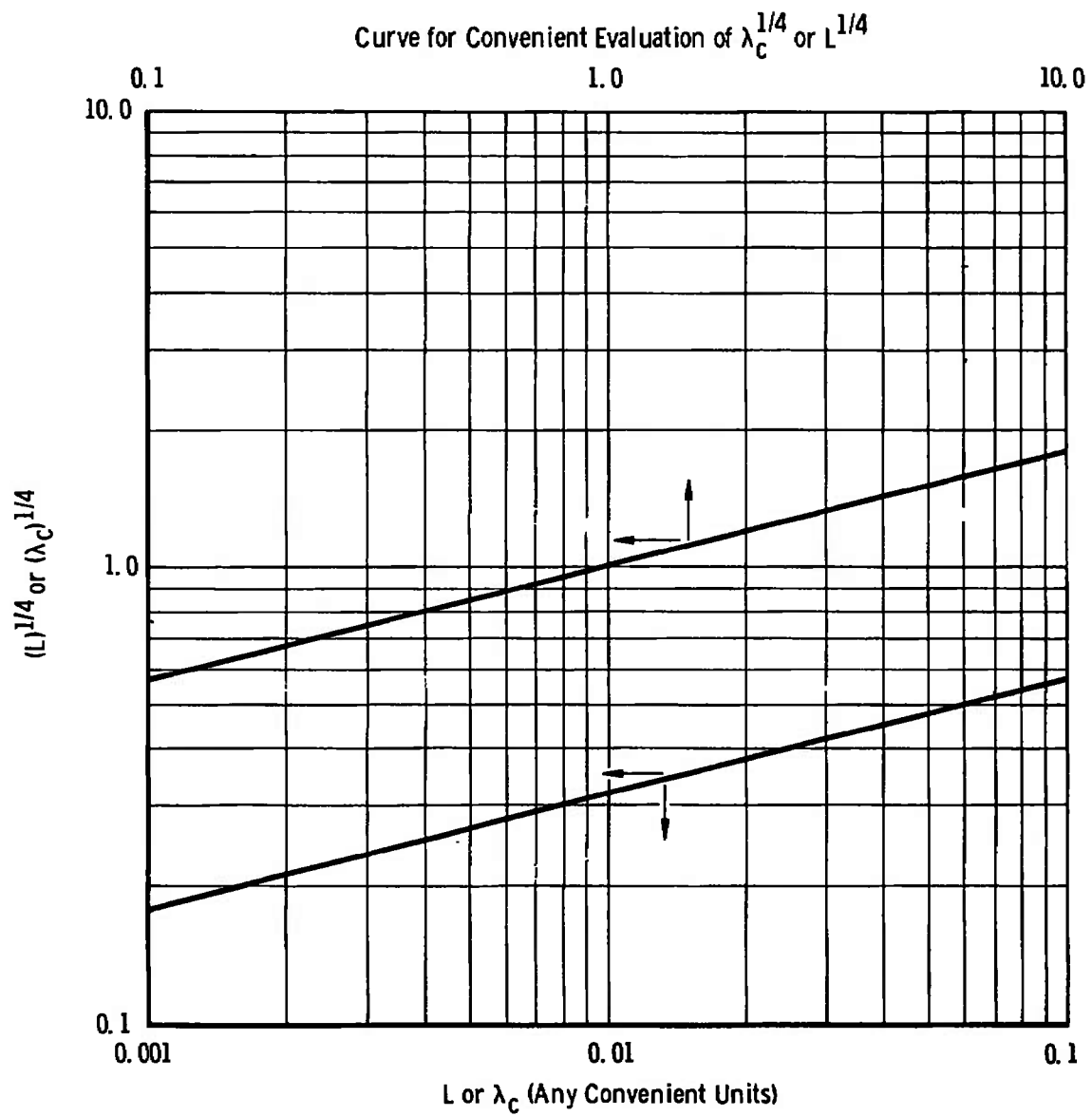
Fig. 16 Plot of Critical Wavelength Parameter or Length Raised to the One-Fourth Power
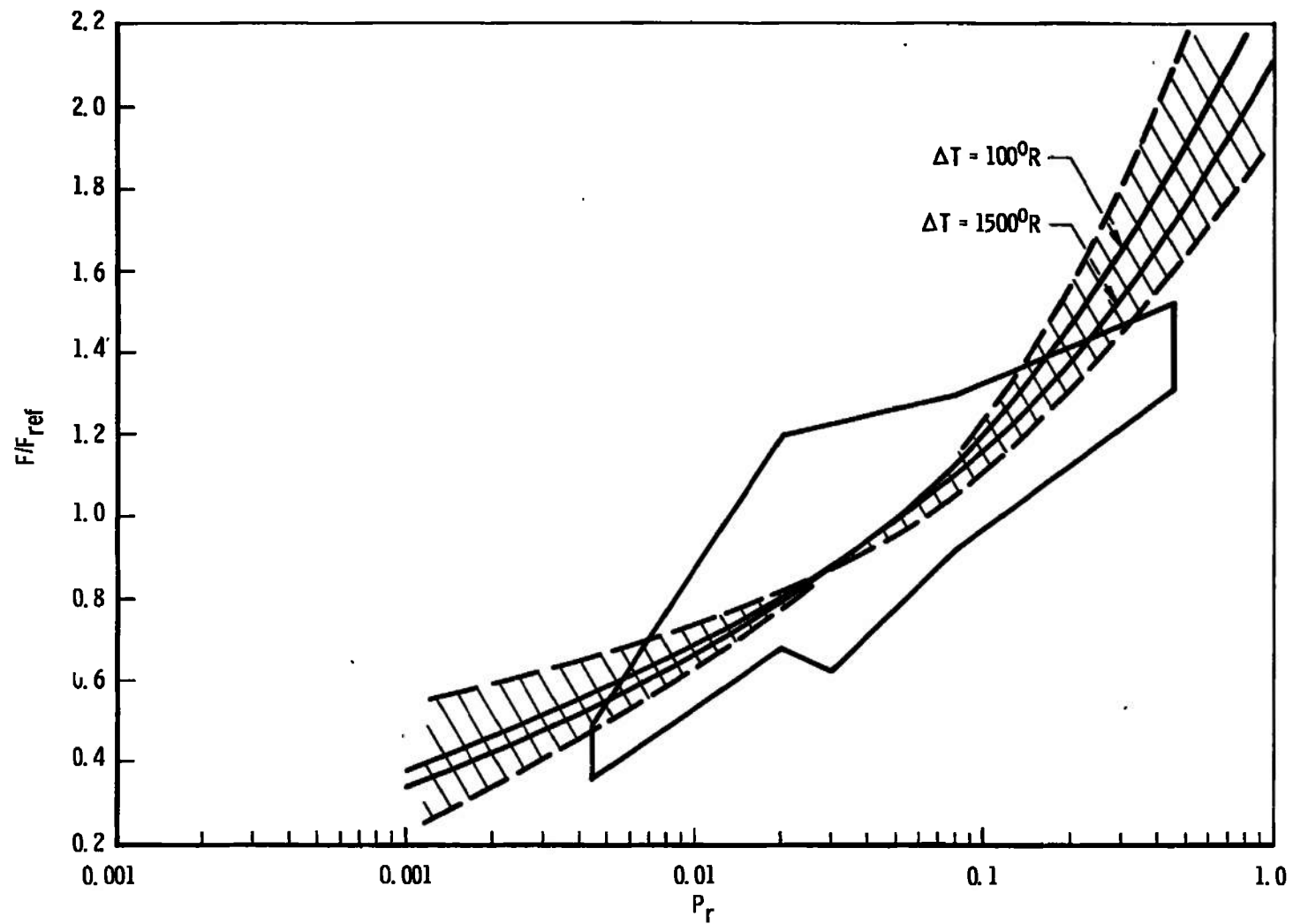
Fig. 17 Comparison of Range of Experimental Data on Film Boiling with Accuracy of the Method Based on the Correspondence Principle
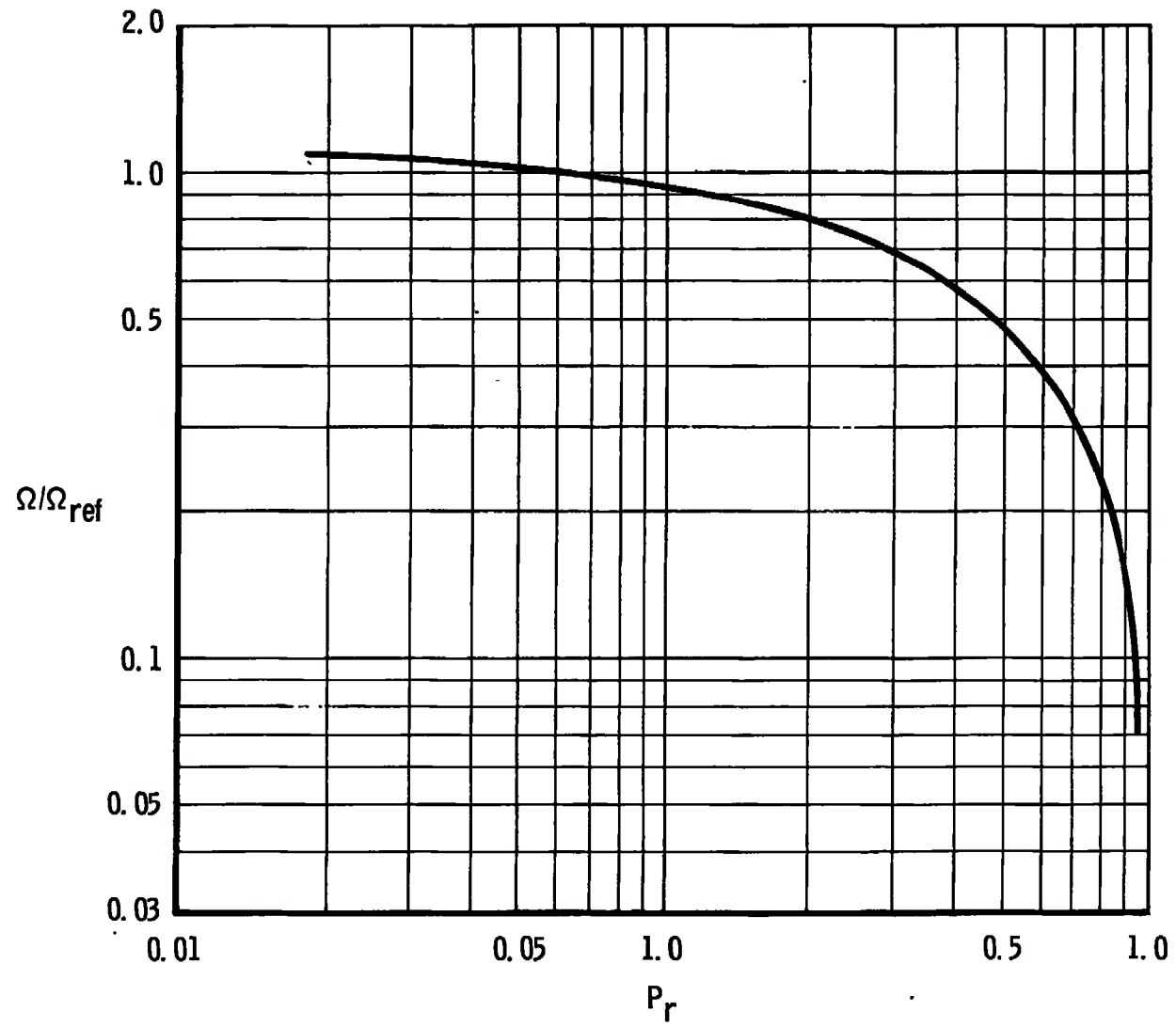
Fig. 18 Minimum Film Boiling Heat Flux Parameter

TABLE I

REFERENCE VALUES OF $(T_w - T_{sat})_{incip}/N_{PR}\sqrt{(q/A)_{incip}}$ EVALUATED AT $P_r = 0.05$

Fluid	X_{ref}, $^\circ R/(Btu/hr\text{-}ft^2)^{1/2}$	Fluid	X_{ref}, $^\circ R/(Btu/hr\text{-}ft^2)^{1/2}$
Ammonia	0.0138	Oxygen	0.0162
Carbon Dioxide	0.0379	Benzene	0.0428
Carbon Tetrachloride	0.0436	Water	0.0141
Mercury	0.0127	Freon 12	0.0453
Neon	0.0077	Ethanol	0.0285
n-pentane	0.0388	Acetone	0.0358
Propane	0.0307	Kerosene (JP-4)	0.0405
Para-hydrogen	0.0108	Helium	0.0074
Nitrogen	0.0166	Argon	0.0192

| Fluid	X_{ref}, $^\circ K/(joule/hr\text{-}m^2)^{1/2} \times 10^{-4}$	Fluid	X_{ref}, $^\circ K/(joule/hr\text{-}m^2)^{1/2} \times 10^{-4}$
Ammonia	0.719	Oxygen	0.845
Carbon dioxide	1.975	Benzene	2.231
Carbon tetrachloride	2.272	Water	0.735
Mercury	0.662	Freon 12	2.361
Neon	0.401	Ethanol	1.486
n-pentane	2.023	Acetone	1.866
Propane	1.600	Kerosene (JP-4)	2.111
Para-hydrogen	0.563	Helium	0.386
Nitrogen	0.865	Argon	1.001

TABLE II
PRANDTL NUMBERS OF SATURATED LIQUIDS VERSUS REDUCED PRESSURE

P_r	Nitrogen	Oxygen	Propane	Water	n-Pentane	Ammonia	Benzene	Carbon Dioxide	Carbon Tetrachloride
0.02	3.02	3.41	2.47	1.22	4.23	2.08	5.20	4.11	3.28
0.05	2.54	2.89	2.64	1.00	3.54	2.04	4.04	3.53	2.57
0.1	2.35	2.45	2.22	0.90	2.91	2.01	3.53	2.77	2.11
0.2	2.25	2.00	1.85	0.87	2.55	1.98	3.04	2.18	1.92
0.4	1.67	1.83	1.94	0.99	2.36	2.05	2.55	2.26	1.74
0.5	1.62	1.91	1.96	1.12	2.30	2.29	2.80	2.43	1.81
0.6	1.59		1.95	1.40	2.22		3.14	2.78	1.90
0.8	1.50	2.25	2.08	2.08	2.15		4.09	4.50	2.39
0.9	1.61	2.46	2.44	6.36	2.06		6.18	8.80	2.63
0.95	1.70		2.88	9.95	2.49		8.34	16.00	3.28

P_r	Ethanol	Freon 12	Helium	Para-Hydrogen	Neon
0.02	9.25	5.08		1.175	5.66
0.05	7.44	3.98	0.453	1.158	3.88
0.1	6.40	3.63	0.444	1.127	2.93
0.2	5.78	3.37	0.488	1.103	2.35
0.4	5.53	3.21	0.575	1.113	1.92
0.5	5.84	3.16		1.129	1.79
0.6	6.35	3.37		1.145	1.73
0.8	11.3	3.56		1.550	
0.9	18.0	5.09			
0.95		5.96			

TABLE III
INCIPIENT BOILING WALL TEMPERATURES FOR LIQUID NITROGEN AND
FOR WATER FLOWING THROUGH A 0.5-IN. ID PIPE

Nitrogen			
Conditions: $\dot{m}$ = 1.0 lb_m/sec T_{in} = 100°R T_{exit} = 140°R P_{in} = 100 psia			
	Based on Inlet Conditions	Based on Average Conditions	Based on Exit Conditions
$(T_w)_{incip}$	181°R	176°R	168°R
$(\Delta T_{sat})_{incip}$	5.3°R	5.1°R	4.4°R
Water			
Conditions: $\dot{m}$ = 1.0 lb_m/sec T_{in} = 100°F T_{exit} = 140°F P_{in} = 100 psia			
	Based on Inlet Conditions	Based on Average Conditions	Based on Exit Conditions
$(T_w)_{incip}$	348°F	331°F	314°F
$(\Delta T_{sat})_{incip}$	20°F	20°F	24°F

TABLE IV

REFERENCE VALUES OF $(T_w - T_{sat})/(q/A)^{0.3}$ EVALUATED AT $P_r = 0.05$

Fluid	$K_{ref} = \Delta T_{sat}/(q/A)^{0.3}$, $°F/(Btu/hr-ft^2)^{0.3}$	$K_{ref} = \Delta T_{sat}/(q/A)^{0.3}$, $°K/(joules/hr-m^2)^{0.3}$
Ammonia	0.727	0.0246
n-pentane	1.375	0.0466
Neon	0.702	0.0238
Freon 12	2.531	0.0858
Helium	0.230	0.0078
Oxygen	1.080	0.0366
Nitrogen	0.913	0.0309
Para-hydrogen	0.487	0.0165
Water	0.628	0.0213
Ethanol	3.397	0.1151
Kerosene (JP-4)	3.892	0.1319
Propane	1.106	0.0375
Carbon Tetrachloride	2.063	0.0699
Carbon dioxide	1.611	0.0546
Argon	1.179	0.0400

TABLE V

REFERENCE VALUES FOR SATURATED POOL BOILING

BURNOUT (EVALUATED AT $P_r = 0.05$ AND $\frac{a}{g} = 1$)

Fluid	$(\phi_{ref})_{min}, \frac{Btu}{hr\text{-}ft^2}$	$(\phi_{ref})_{max}, \frac{Btu}{hr\text{-}ft^2}$
Ammonia	373, 300	488, 100
Argon	78, 900	103, 100
Carbon dioxide	153, 100	200, 200
Carbon tetrachloride	81, 400	106, 400
Ethanol	226, 000	294, 900
Freon 12	84, 400	110, 400
Helium	1100	1430
Para-hydrogen	22, 300	29, 200
Kerosene (JP-4)	98, 400	128, 700
Mercury	881, 000	1, 151, 000
Neon	33, 400	43, 700
Nitrogen	59, 000	77, 200
Oxygen	94, 200	123, 200
n-pentane	86, 200	112, 800
Propane	67, 900	88, 700
Water	850, 500	1, 112, 000

TABLE VI

REFERENCE VALUES FOR BURNOUT SUBCOOLING EFFECT

$$\psi_{ref} = \frac{(q/A)}{C_\ell \Delta T_{sub}} \text{ AT THE REFERENCE REDUCED PRESSURE OF } P_r = 0.05$$

Fluid	$(\psi_{ref})_{min}$, $lb_m/hr\text{-}ft^2$	$(\psi_{ref})_{max}$, $lb_m/hr\text{-}ft^2$
Ammonia	2810	3680
Argon	3650	4770
Carbon dioxide	4150	5430
Carbon tetrachloride	4520	5910
Ethanol	2630	3440
Freon 12	4180	5470
Helium	216	283
Para-hydrogen	224	292
Kerosene (JP-4)	2200	2870
Mercury	80, 700	106, 000
Neon	2340	3060
Nitrogen	2300	3010
Oxygen	2830	3690
n-pentane	2090	2740
Propane	2970	3880
Water	4280	5590

TABLE VII

CORRECTION FACTOR FOR SURFACE ORIENTATION

Direction of Buoyancy Force	Multiplicative Factor, b
Away from the surface	1. 00
Toward the surface	0. 50
Parallel with the surface	0. 75

TABLE VIII
REFERENCE VALUES OF F_{ref}

ΔT	F_{ref}				λ_{ref}
	100	500	1000	1500	
Ammonia	41.10	33.75	43.38	X	0.04106
Freon	46.34	35.26	40.99	X	0.02034
Helium	16.94	26.86	33.99	41.72	0.00887
Para-hydrogen	20.39	31.36	41.72	50.20	0.03745
Nitrogen	16.93	15.87	16.71	17.93	0.02076
Neon	23.25	20.50	23.12	X	0.01265
Oxygen	20.18	18.56	19.66	21.71	0.02087
n-pentane	43.2	37.02	41.42	47.3	0.02800
Water	60.92	44.02	41.10	47.93	0.04497
Argon	16.13	13.47	13.62	14.04	0.01818
Carbon dioxide	17.90	16.72	20.27	22.77	0.02376
Propane	24.98	20.42	23.23	27.26	0.03194

TABLE IX
REFERENCE VALUES OF $(q/A)_{MC}/(\rho v)_{film}$ EVALUATED AT $P_r = 0.05$

Fluid	$(\Omega_{ref})_{max}$, Btu-lb$_m$/hr-ft^5	$(\Omega_{ref})_{min}$, Btu-lb$_m$/hr-ft^5
Ammonia	3679	2813
Argon	4773	3650
Carbon dioxide	5430	4152
Ethanol	3435	2627
Freon 12	5467	4180
Helium	283	216
Para-hydrogen	292	224
Kerosene (JP-4)	2874	2198
Mercury	105,500	80,690
Neon	3061	2341
Nitrogen	3009	2301
Oxygen	3694	2825
n-pentane	2736	2092
Propane	3883	2969
Water	5593	4277

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13. ABSTRACT

Selected boiling heat-transfer correlations currently being used for design purposes are presented in a graphical form which enables the engineer to compute numerical values of the boiling heat flux parameters rapidly and without knowledge of the physical properties of the fluid other than at one arbitrary reference state.

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14. KEY WORDS	LINK A		LINK B		LINK C	
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